PRODUCT ENGINEERING

VOLUME 7 NUMBER 9

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October Number

For many designing engineers, the greatest event in October will be the Nationals Metals Congress and Exposition. Whether or not you attend this convention and exposition, you will be interested in reading in October *P.E.* a summary of the exhibits to be seen, with brief descriptions of many of the feature displays. Those who will attend the show will thus be able to plan their visits to be sure to see what interests them most. Those who do not attend will "see the show in print."

Brief abstracts of papers to be delivered at the technical sessions will also be published in October *P.E.* It will enable readers to pick their meetings with a better knowledge of what they will hear. And, of course, it will enable them to select the papers which they want to get in printed form.

In addition to this, October *P.E.* will have general articles. Dr. MacQuigg of Union Cabide and Carbon Company will tell about design factors in using the moderate alloy high-strength steels. "Aluminum Hot Pressings," and "Mistakes to Avoid in Specifying Steels" will be the subject of two other articles. And, of course, there will be the usual departments and reference book sheets.

Coming

November *P.E.*—The designing engineers' reference number.

Report on activity and trends in product design.

A resume of new and improved parts and materials.

A comprehensive "Where-To-Buy Directory" covering parts, materials and finishes; completely classified and cross indexed, with explanatory notations.

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CONTENTS FOR SEPTEMBER, 1936

TESTS OF GEAR AND HYDRAULIC DRIVES 326
DR. ENG. M. KRONENBERG—An analysis of the limitations and advantages inherent in the design of hydraulic versus gear drives.

HOW TO MAKE ALIGNMENT CHARTS—VII 334
M. G. VAN VOORHIS—Last of a series of articles showing how to make nomograms for the solution of equations.

ANALYSIS AND COMPARISON OF CAM ACTIONS 339
R. E. GROBE—Investigation of three profiles for points of shock by analysis of displacement, velocity and acceleration curves.

SLIDER-CRANK MECHANISMS 341
J. L. SWYERS—Simplifying calculations for determination of piston acceleration and inertia torque of reciprocating mechanism.

PROPORTIONING SIZES OF SIMILAR PARTS 345
CYRIL DONALDSON—Discussion of constant and variable characteristics when designing by "scaling up" or "scaling down" an old machine.

BAFFLING NOISE AND PROVIDING VENTILATION 348
S. C. EAKER—Development of ducts and baffles in an electric refrigerator silencing system with tests for relative damping effects.

TYPICAL JOINTS OF ARC-WELDED STRUCTURALS—II 354
R. A. GAST—Remainder of 44 various arc-welded joints for fabrication of beams, angles and channels.

SLIDER-CRANK MECHANISM 363
J. L. SWYERS—Factors for determining piston acceleration and inertia torque in slider-crank mechanisms.

Editorial 325
Product Developments 331
Question and Comment 351
News 356
Meetings 357
New Materials and Parts 358
Manufacturers' Publications 361
Books and Bulletins 362

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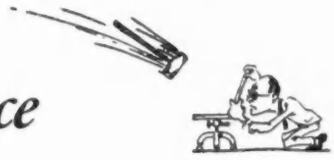


What's Wrong Here?

PRODUCT ENGINEERING
NEW YORK, N. Y.

In as much as we have had considerable trouble with cadmium plated springs due to the increased brittleness caused by the cadmium plating, we are especially interested in the paragraph that states that steel springs may have their endurance limits increased by as much as 100 per cent by an electroplated coating of cadmium 0.0002 in. thick. This is on page 317 of the August issue of *Product Engineering* and is itemized as paragraph No. 26.

Any information that you may give us will be greatly appreciated. —H.A.R.
Toledo, Ohio

THE item referred to was abstracted from the publication of a spring manufacturer. Electroplating is a "tricky" process if not properly controlled and it is possible that others have had experiences similar to that of H.A.R. The question is, does the brittleness of cadmium plating decrease the fatigue strength of such electroplated springs? We will be glad to hear from others.

PRODUCT ENGINEERING
NEW YORK, N. Y.

The August number of *Product Engineering* arrived. It is usually 100 per cent right, but the August number is a mess and shows the effects of the vacation season on your binding department. In my copy, pages 294 to 300 are duplicated while pages 301 to 316 are missing.

I know mistakes will happen and cannot always be avoided. No offense is intended in this letter. If you could send me pages 301 to 316 it would be greatly appreciated. I want to keep my files complete. —C.E.K.
Watertown, N.Y.

FRANKLY, the editor had a sinking feeling until he read as far as "on your binding department." When it was evident that the mistake was a mechanical one that could readily be rectified, he felt better.

But C.E.K. made two mistakes. In his anxiety to be sure to get his files complete, he inclosed postage. Secondly, he asked for only the missing pages. Every subscriber is entitled to complete copies, so we sent C.E.K. another copy—and the stamps which he had sent.

Once in a while it happens that the binding machine "goes haywire." The operator is supposed to check possible mistakes, but they do slip by occasionally. Of course it was gratifying to know that C.E.K. attaches so much value to his files of P.E., but we are glad to have had the opportunity of rectifying the binding error. A postal card will do, if you ever get a mutilated or improperly bound copy.

The Coming Index

BECAUSE of the expressed desire of many of our readers, November *Product Engineering* will include a comprehensive directory of manufacturers of parts, materials, finishes and engineering department equipment.

In addition to having detailed classifications and cross references, there will be explanatory notations that will make the directory more specific. More than 500 manufacturers will be listed. Cooperation from our readers to make this directory as comprehensive as possible, will be appreciated.

Nomographing the Babies

PRODUCT ENGINEERING
NEW YORK, N. Y.

Please send us copies of all the articles "How to Make Alignment Charts" by M. G. Van Voorhis that have appeared in recent numbers of *Product Engineering*. Send the bill to me at the Babies Hospital, New York, N. Y. —Dr.O.S.W.
New York, N. Y.

A QUEER place, we thought, from which to get a request for such articles. A telephone call to the doctor cleared up the mystery. No, it is not that mathematics is a hobby of the doctor. They take numerous measurements of babies and then make certain calculations to determine various biological, physiological, babological or whatever they call them, factors. Nomograms are desired to eliminate the arithmetic.

And Still They Come

PRODUCT ENGINEERING
NEW YORK, N. Y.

Will you please send me reprints of the series of four articles on "Stress Calculations for Thin Aluminum Alloy Sections" by S. A. Kilpatrick and O. J. Schaefer, which appeared in the February, March, April and May numbers of *Product Engineering*, and bill me for them. —D.B.H.
Pittsburgh, Pa.

WE did the best we could, but are now all out of tear sheets and extra copies of the issue. The only way we could meet such requests would be to reprint the whole series in booklet form to sell at 50 cents a copy. We will be glad to do this if there is a sufficient demand. Here's the last call. All who may want to get such a booklet, please place your order now, to be delivered, "if and when issued."

Loose Leaf Binders?

PRODUCT ENGINEERING
NEW YORK, N. Y.

Being one of the first subscribers to *Product Engineering*, I have been filing the reference sheets and also some of the articles. For this reason I would like to get a loose leaf binder for the reference book sheets and articles. Do you have such a binder suitably embossed that can be obtained as a special offer? —J.F.S.
Philadelphia, Pa.

WE do not at present have a binder such as J.E.S. suggests. But if sufficient readers are really interested in procuring such a binder the editor might be able to induce the publisher to make one available. Let all who are interested speak now. A postcard will be sufficient.

Why Elliptical Gears?

PRODUCT ENGINEERING
NEW YORK, N. Y.

Recently, in the development of an automatic machine, I faced the necessity of designing elliptical gears. Such gears were something new in my practical experience and I turned to the available literature for specific information. After a long search, I succeeded in finding some information but little or none in usable form both from the practical and theoretical angles.

In spite of all difficulties, the gears were developed and now I find that such gears have unique advantages. They not only make possible unusual speed relationships and timing, but when used in conjunction with linkage give some very unusual motions. Why is it that there does not seem to be much general interest in the use of such gears? And why is there so little published data about them? Has *Product Engineering* ever printed any articles on this subject? —P.J.J.
New York, N. Y.

THE answer to the last question is no, for the reason that we also felt that there was little general interest in the subject. But maybe we are wrong. There are many evident possibilities in elliptical gears and perhaps some of our readers have thoughts on the subject that they would like to pass along.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. SEPTEMBER 1936 VOLUME SEVEN, NUMBER NINE

A Broader Outlook

AS AMERICA emerges from depression, courage and enterprise regain their old position in American character, and outlooks broaden. It is true in politics, where both major parties proclaim their liberalism, and the nuisance party bellows its own exclusive right to that designation. It is true in industry, where wide plans for expansion are taking shape, and money is being appropriated for modern equipment and better buildings. It is most particularly true of engineering and design, where new ideas are being sought and traditional notions are being challenged.

Broader outlooks on engineering matters seldom start in brains stunted by thinking too much about details in a single field of endeavor. Too much specialization is deadly to broadening vision, particularly if that specialization is in a field that is being narrowed by competing products.

Turn the page a moment and glance at the article that begins there. It analyzes the results of tests made to show the relative capabilities of mechanical and hydraulic drives for machine tools. Breadth of vision is essential to an adequate estimate of the merits of each type of drive. An engineer who had spent years on the mechanical drive with no attention to hydraulics would be handicapped in attempting comparisons, because his subconscious tendency would be to favor the functions with which he was familiar, and to doubt those which were foreign to his experience.

Another example of the current demand for broader outlooks is in the field of railroad car design. For many years both freight and passenger cars have been strengthened to stand the heavier duty imposed by speeds and loads, by adding more metal to sections. Heavier cars meant heavier and more powerful engines, and these led to a general strengthening of bridges. A vicious circle was started in which pay load became a smaller and smaller proportion of the total, and profits dwindled. Engineers trained in the railroad business found it hard to break out of the circle. Companies that had the courage to try the strong alloys to get dead weights down, and to apply automobile and airplane experience to railroad vehicles were unable to get enough designers with basic railroad training and familiarity with the progress in other transport fields. Men who know only one branch or the other did not suffice.

Experiences like this one have been common in industry. Possibly there should be a change in our engineering department organizations. Certainly there should be constant effort by the chief engineer to discover subordinates with potentially broadening outlooks, and to develop such men as rapidly as circumstances permit. We can't have too many of them.

Comparative Test Results of Gear and Hydraulic Drives

DR. ENG. M. KRONENBERG

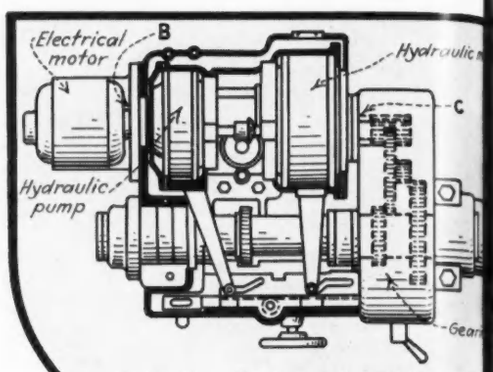


FIG. 1 Rack operated sliding cam to control eccentricity of pump and hydraulic motor

TO DETERMINE the efficiency, power consumption and cutting capacity of a hydraulically-driven lathe versus a mechanically-driven one, two machines each of 20 in. swing were chosen and tested.

The hydraulically-driven lathe was equipped with a Lauf-Thoma drive, shown in Fig. 1, which is a combination unit powered by a constant speed electric motor directly connected to a variable stroke rotary plunger pump with the discharge piped to the input side of a variable stroke rotary plunger hydraulic motor. The length of plunger stroke in both pump and motor can be varied by changing the eccentricity of a carrier element. The eccentric positioning controls are so arranged that the hydraulic motor runs at full stroke (greatest eccentricity) while the pump eccentricity changes from zero to its maximum. After the point of maximum eccentricity of pump is reached the hydraulic motor eccentricity can be changed from its maximum to zero. The speed of the hydraulic motor therefore depends upon either its own eccentricity or the eccentricity of the pump. The hydraulic motor drives the lathe spindle through back gears.

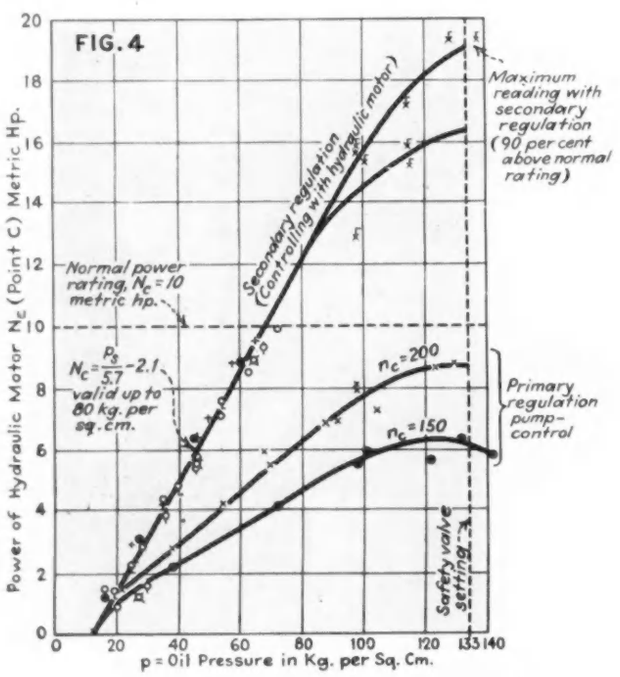


Fig. 4—Power of hydraulic motor at C and corresponding oil pressure when controlling speed with pump and with hydraulic motor

The hydraulic lathe spindle had a stepless speed range from 0 to 600 r.p.m., the spindle speed of the gear driven lathe ranged from 9.6 to 480 r.p.m. in geometrical steps with a 1.26 ratio. When testing, the energy input was measured with a wattmeter and checked with an ammeter and a voltmeter. The outgoing energy was measured with a prony-brake, as shown in Fig. 2; it was also measured when cutting with a cutting-pressure indicating device shown in Fig. 3. The driving motors and the hydraulic drive were also tested with a brake as separate units.

In this discussion the following notations are used:

B = place where energy enters the hydraulic lathe
= point C of entrance of energy in the gearing of the hydraulic lathe
D = point of exit of energy from the hydraulic lathe (face plate)
C_{ks} = cutting pressure in kg. per sq.mm. of chip area
d = turning diameter
e_p = eccentricity in mm. of the hydraulic pump
e_s = eccentricity in mm. of the hydraulic motor
F = area of chip in sq. mm.
K = a constant for particular machine
M_p = displacement in c. c. per sec. of the hydraulic pump
M_s = displacement in c. c. per sec. of the hydraulic motor
n_B = r.p.m. at B (motor shaft)
n_C = r.p.m. at C (driven-shaft)
n_D = r.p.m. at D (face plate) hydraulically driven machine
N_B = Hp. (75 kg.m. per sec.) at B
N_C = Hp. at C
N_D = Hp. at D
p = oil pressure in kg. per sq. cm.
p_p = oil pressure when controlling speed with pump
p_s = oil pressure when controlling speed with hydraulic motor
S = slip in per cent
S_1 = slip-factor
v = cutting speed in m. per min.
η_F = efficiency of hydraulic drive from B to D
η_{FO} = efficiency of hydraulic drive from B to C
η_R = efficiency of mechanical drive from B_R to D_R
η_Z = efficiency of the gearing of the hydraulic drive from C to D
ρ_{NB} = chip volume per min. per hp. in c. c.
τ_{NB} = chip weight per min. per hp. in kg.

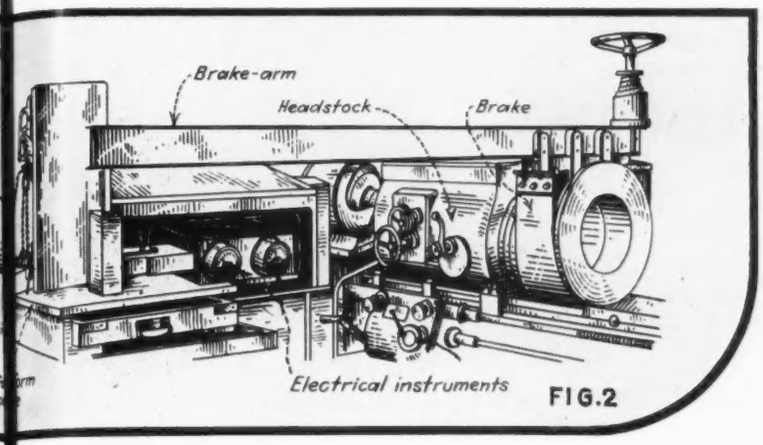


FIG.2

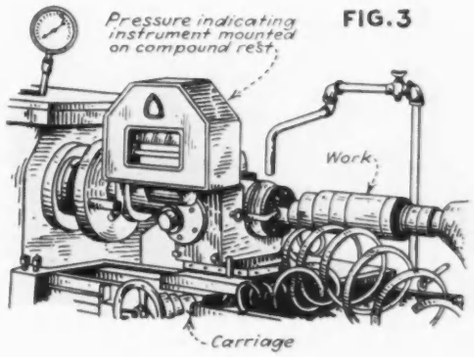


FIG.3

Fig. 1—Headstock of lathe equipped with Lauf-Thoma hydraulic drive showing sliding cam for controlling eccentricity. Fig. 2—Prony-brake arrangement for obtaining energy output of hydraulically operated lathe. Fig. 3—Device for indicating the pressure of cutting tool on work while turning

For a pump-motor combination drive the speed of the hydraulic motor may be expressed as

$$n_C = K e_p / e_s \text{ r.p.m.} \quad (1)$$

which shows that the number of revolutions of the hydraulic motor increase with an increasing eccentricity of the pump and a decreasing eccentricity of the hydraulic motor. By changing the pump eccentricity a regulation of motor speed was obtained between 150 r.p.m. minimum and 350 r.p.m. maximum. By changing the motor eccentricity e_s with the pump at a constant displacement e_p, a speed regulation of the motor was obtained between 350 r.p.m. minimum and 1,200 r.p.m. maximum.

The speeds of the main spindle with a 1 to 16 ratio of back gearing were $n_b = 9.4$ to 75 r.p.m., and with a 1 to 2 ratio of back gearing $n_b = 75$ to 600 r.p.m.

The theoretical horsepower N_c of the Lauf-Thoma drive can be derived from the expression:

$$N_C = \frac{M_s p_s}{75 \times 100} \quad (2)$$

and from the specifications of the unit

$$M_s = 94.5\, e_p$$

and the oil pressure

$$p = (80\, N_c)/e_p \text{ kg. per sq. cm.} \quad (3)$$

Equation (3) shows that the oil pressure is influenced only by the eccentricity of the pump e_p provided the load N_c is constant. The oil pressure can be varied between

$$p_p = 13.3\, N_C \text{ and } 5.7\, N_C \text{ kg. per sq. cm.} \quad (4)$$

When the eccentricity of the hydraulic motor is changed and e_p remains constant the oil pressure is constant for constant loads:

$$p_s = 5.7\, N_C \text{ kg. per sq. cm.} \quad (5)$$

The Lauf-Thoma drive was designed for a rating of 10 metric hp. at the point C (driven shaft). When the eccentricity of the motor controls the speed the oil-pressure should theoretically be 57 kg. per sq. cm., while for the case of pump, the oil-pressure increases according to Equation (4) from 57 kg. per sq. cm. up to 133 kg. per sq. cm.

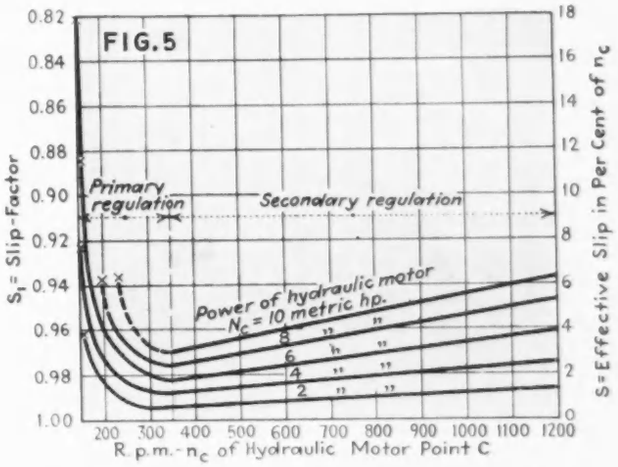


Fig. 5—Slip-curves of Lauf-Thoma-drive, slip of electric motor not included, shown as a function of load on hydraulic motor

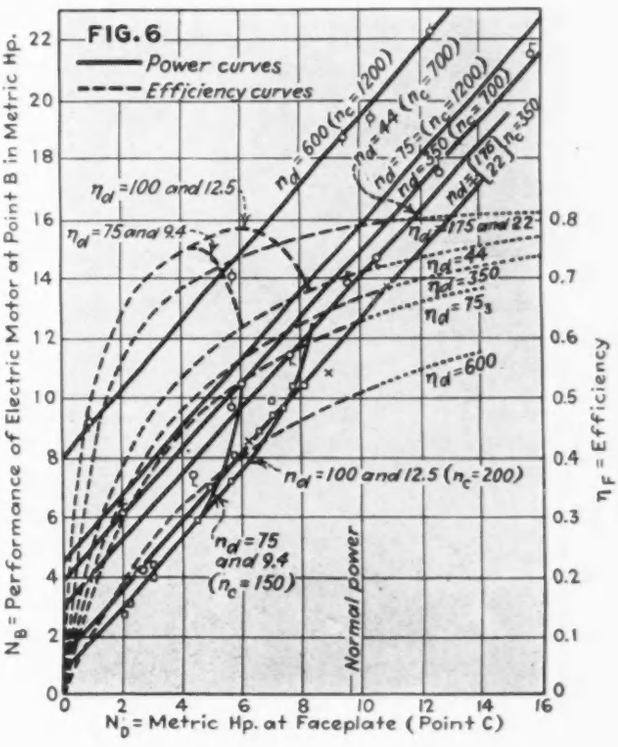


FIG.6
Power curves
Efficiency curves
n_d = 600 (n_c = 1200)
n_d = 44 (n_c = 700)
n_d = 75 = (n_c = 1200)
n_d = 350 (n_c = 700)
n_d = 175 and 22 (n_c = 350)
n_d = 100 and 12.5
n_d = 75 and 9.4
n_d = 175 and 22
n_d = 44
n_d = 350
n_d = 75
n_d = 600
n_d = 100 and 12.5 (n_c = 200)
n_d = 75 and 9.4 (n_c = 150)
Normal power
N_B = Performance of Electric Motor at Point B in Metric Hp.
N_D = Metric Hp. at Faceplate (Point C)
η_F = Efficiency

Fig. 6—Power input and output curves with efficiencies for the hydraulic drive (gearing included)

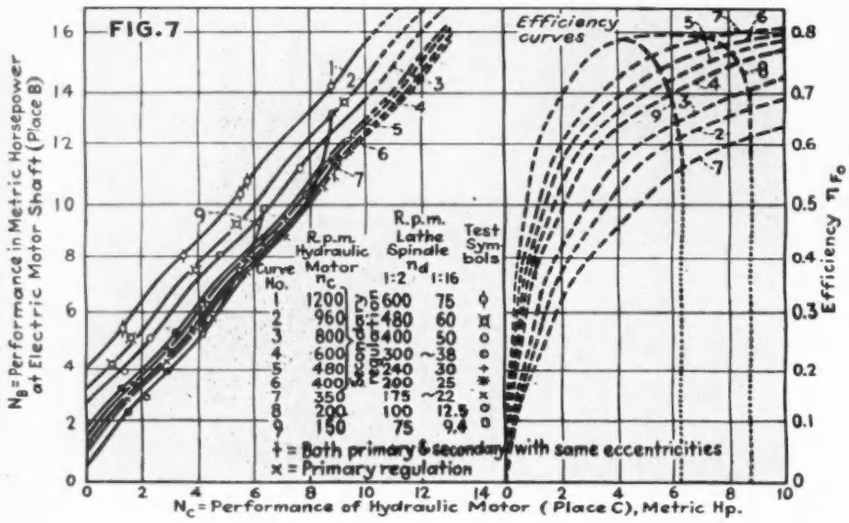
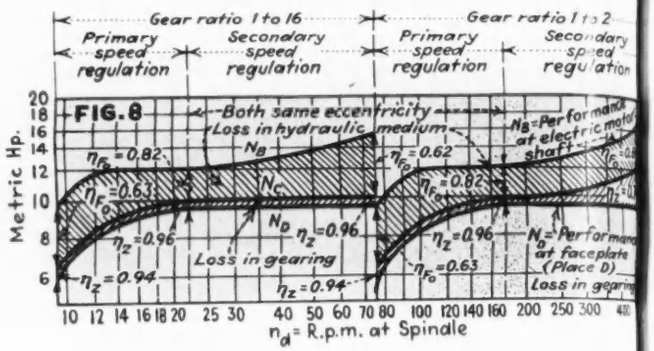


FIG. 7
Efficiency curves
N_B = Performance in Metric Horsepower at Electric Motor Shaft (Place B)
N_C = Performance of Hydraulic Motor (Place C), Metric Hp.
Efficiency η_{Fo}
Curve No. | R.p.m. Hydraulic Motor n_C | R.p.m. Lathe Spindle n_d 1:2 | 1:16 | Test Symbols
1 | 1200 | 600 | 75 | ϕ
2 | 960 | 480 | 60 | ¤
3 | 800 | 400 | 50 | o
4 | 600 | 300 | ~38 | o
5 | 480 | 240 | 30 | +
6 | 400 | 200 | 25 | *
7 | 350 | 175 | ~22 | x
8 | 200 | 100 | 12.5 | o
9 | 150 | 75 | 9.4 | □
+ = Both primary & secondary with same eccentricities
x = Primary regulation



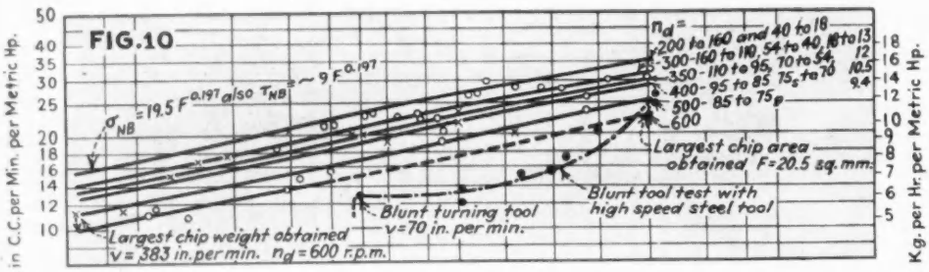


FIG. 10
$\sigma_{NB} = 19.5\ F^{0.197}$ also $\tau_{NB} = 9\ F^{0.197}$
n_d =
200 to 160 and 40 to 18
300-160 to 110, 54 to 40, 18 to 13
350-110 to 95, 70 to 54, 12
400-95 to 85, 75 to 70, 10.5
500-85 to 75, 9.4
600
Largest chip area obtained F=20.5 sq. mm.
Blunt tool test with high speed steel tool
Blunt turning tool v=70 in. per min.
Largest chip weight obtained v=383 in. per min. n_d=600 r.p.m.
Volume of Chips in C.C. per Min. per Metric Hp.
τ_{NB} = Weight of Chips in Kg. per Hr. per Metric Hp.

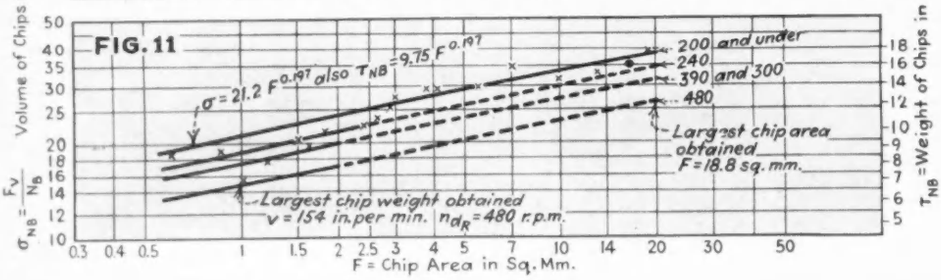
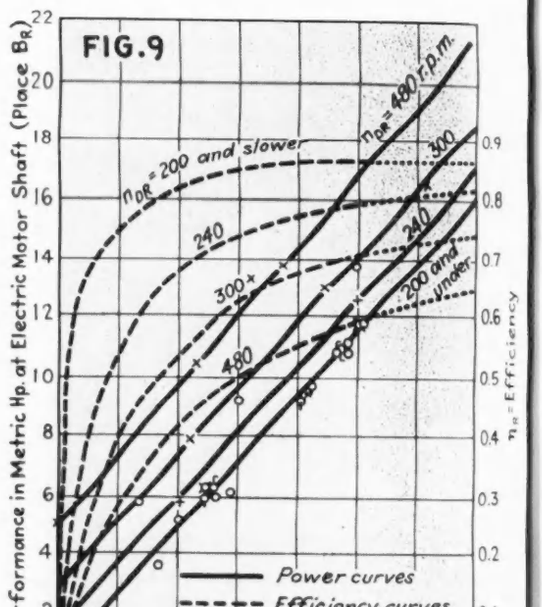


FIG. 11
$\sigma = 21.2\ F^{0.197}$ also $\tau_{NB} = 9.75\ F^{0.197}$
200 and under
240
390 and 300
480
Largest chip area obtained F=18.8 sq. mm.
Largest chip weight obtained v=154 in. per min. n_{dR}=480 r.p.m.
$\sigma_{NB} = \frac{F_v}{N_B}$
F = Chip Area in Sq. Mm.



With the safety valve set to relieve at 133 kg. per sq.cm. these theoretical derivations make clear that the Lauf-Thoma drive can be greatly overloaded in the case of motor control, as a load of 10 metric hp. causes an oil pressure of only 57 kg. per sq.cm. In the case of pump control, however, a load of 10 metric hp. is only obtainable if no slip occurs.

These deductions were checked by braking tests on the hydraulic lathe, especially the relations between speed, oil pressure and load. The maximum load on the drive was found to be 19 metric hp., a 90 per cent overload under which the machine ran without any difficulty. The temperature of the oil was 40 deg. C. As will be apparent from the left hand straight line in Fig. 4, the oil pressure rose only up to 69 kg. per sq. mm. for full-load of 10 metric hp. in the case of motor control, or 12 kg. per sq. mm. higher than the theoretical value of 57 kg. per sq. mm. This difference is the idle running loss of the drive. Equation (5) when corrected to agree with the test results is written

$$p_s = (5.7\ N_C + 12) \text{ kg. per sq. mm.}$$
$$\text{or } N_C = (p_s\ 5.7 - 2.1) \text{ metric hp.} \qquad (6)$$

The tests show this equation does not hold for loads over 80 kg. per sq. mm.

In the case of pump control the behavior of the Lauf-Thoma drive is quite different. The attainable load N_C increases (see Fig. 4) at a lower rate with the oil pressure than in the case of motor-control. The maximum permissible oil pressure of 133 kg. per sq. cm. for n_c = 200 r.p.m. was obtained by a load of 8.8 metric hp. and for n_c = 150 r.p.m. by a load of 6.3 metric hp.

Fig. 5 shows curves for the slip at different speeds and loads. The smallest slip was found for n_c = 350 r.p.m. where the eccentricity of pump is the same as that of hydraulic motor. The slip increases rapidly when the speed is less than 200 r.p.m. The curves do not include the slip of the electric driving motor but only the slip of the oil gear.

Let S be the slip, measured in per cent so that the slip factor

$$S_1 = 1 - (S/100) \qquad (7)$$

or the value with which n_c is to be multiplied in order to get its effective revolutions. Fig. 6 shows the results of the investigations of the complete headstock, and

Fig. 7 those of the tests taken without the back gearings, that is, only the Lauf-Thoma drive.

For the complete head-stock the efficiency was found to be 82 per cent when overloaded with the eccentricity of pump and hydraulic motor the same and $n_c = 350$ r.p.m.; $n_D = 22$ r.p.m. and 175 r.p.m. For normal load n_D of 9.6 metric hp. at the face plate the efficiency was 78 per cent, and $n_c = 350$ r.p.m., $n_D = 22$ r.p.m. and 175 r.p.m. For normal load N_c of 10 metric hp. without back gearing the efficiency was found to be 81 per cent, and in the case of overloading was 84 per cent efficient.

It is interesting to observe the efficiency with pump-control, although the full rated load could not be reached, the efficiency went up to 80 per cent (without gearing) and 78.5 per cent (with gearing) at loads of 5 and 7 metric hp. respectively. The efficiency is always good for small loads with the pump controlling.

A comparison of Fig. 6 and 7 shows the losses through the back-gearing and for high speeds where n_D ranges from 300 to 600 r.p.m. The efficiency of the gearing was found to be 95.5 per cent at pitch-circle speeds less than 2 m. per sec., 90 per cent at 3 m. per sec., and 80 per cent at a speed of 6 m. per sec. and up.

In Fig. 8 the capacity has been analyzed by plotting the values of the horsepower on a bilogarithmic system as a function of the number of revolutions n_D, and shows the variation in the efficiency with r.p.m. for the normal load of 9.6 metric hp. separately for the hydraulic drive and the back-gears.

Fig. 7—Power input and output curves, also efficiency curves for the hydraulic drive, gearing excluded. Fig. 8—Variations in efficiency and capacity at the head-stock with Lauf-Thoma-Drive for full load at the face plate. Fig. 9—Energy input and output curves with efficiency of mechanical lathe. Fig. 10—Chip removing capacity as a function of chip area at various speeds, also curve showing test with blunt tool, for hydraulic lathe. Fig. 11—Same as data in Fig. 10 except that these curves are for mechanical lathe

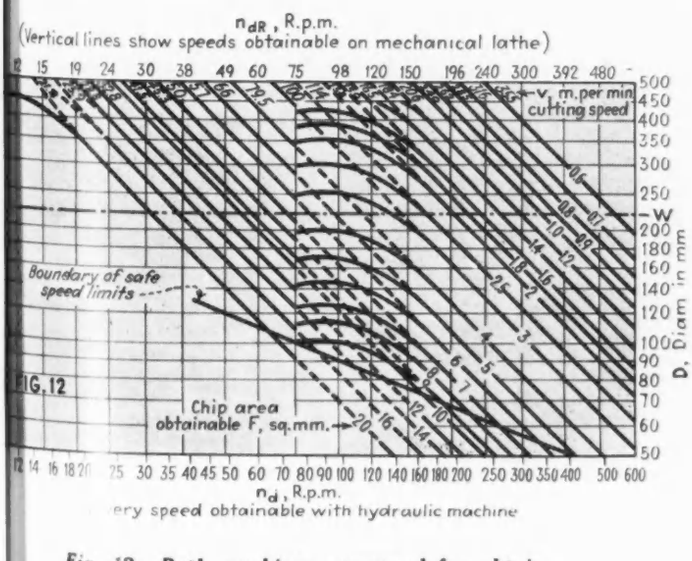

n_{dR}, R.p.m.
(Vertical lines show speeds obtainable on mechanical lathe)
12 15 19 24 30 38 49 60 75 98 120 150 196 240 300 392 480
v, m. per min cutting speed
500 450 400 350 300 250 W 200 180 160 140 120 100 90 80 70 60 50
D, Diam in mm
Boundary of safe speed limits
Chip area obtainable F, sq.mm.
12 14 16 18 20 25 30 35 40 45 50 60 70 80 90 100 120 140 160 180 200 250 300 350 400 500 600
n_d, R.p.m.
very speed obtainable with hydraulic machine
FIG. 12

Fig. 12—Both machines compared for obtainable chip areas and cutting speeds for different diameters of work as a function of the speed

The horsepowers at the points B, C and D of the machine are also indicated. A speed of 75 r.p.m. is obtainable by pump-control with a back gearing ratio of 1 to 2, and also by motor-control with a back-gearing ratio of 1 to 16 so that it is possible by using the motor-control to utilize full driving power and to avoid the unfavorable condition which arises for a speed of 75 r.p.m. when the speed of the spindle is controlled entirely by the pump displacement.

Braking tests on the mechanical lathe also showed that the losses through the gearing are greater for the higher pitch circle speeds than for the lower ones. Fig. 9 shows the efficiency of the head-stock of the mechanical lathe. The difference between the efficiency of the hydraulic and the mechanical drive are very nearly equal at the higher speeds and normal load, the difference in favor of the mechanical lathe is 1.7 per cent at 480 r.p.m., 1.6 per cent at 392 r.p.m., and only 1.4 per cent at 300 r.p.m. At less than 200 r.p.m. the efficiency of the mechanical lathe is somewhat better, 85 per cent against 60 and 78 per cent.

Cutting pressure tests were made on various materials. The cutting pressure increased in proportion to the 0.8th power of the chip area. The specific cutting pressure for an area of chip of 1 sq. mm. was found to be 180 kg. per sq. mm. This was decreased to 100 kg. per sq. mm. when the area of chip was 20 sq. mm., a 44 per cent decrease.

According to metal cutting laws, for the same hp. the chip-volume per min. increases with the area of chip. Nevertheless it is not advisable to make a too extensive use of this increase, because of the effects of the higher total cutting pressure on the machine, on the tools and on the work. The decrease in energy needed as a result of the decrease in specific cutting pressure for larger areas of chip may be utilized to increase the cutting speed.

The values for the area of chip and for the cutting speed were chosen in such relations that it was possible always to obtain full load on the machines. On the hydraulic lathe the smallest area of chip was 0.32 sq. mm. with 600 r.p.m., corresponding to a cutting speed of 383 m. per min. This was less than full load but it was decided to avoid possible accidents by not using higher cutting speeds. The largest area of chip was 20.5 sq. mm. Tungsten-carbide tools were used for cutting chips up to 7 sq. mm. area with corresponding high cutting speeds while high-speed tools were used for the larger chips and slower cutting speeds. In addition tests with high speed tools were conducted to determine the influence of blunting the tool on the energy required and to test the quality of the tools.

The results of the cutting tests are shown in Fig. 10 for the hydraulic lathe, and in Fig. 11 for the mechanical lathe. The chip-volume per min. increases with the area of chip, provided there is a constant load, the specific chip-volume per hp. min. also increases. The influence of the variation in efficiency with speed may be observed from the parallel lines. The influence of the gradual blunting of the high speed tool is apparent from the blunting-curve in Fig. 10. The specific chip volume follows a straight line in the bilogarithmic system as a function of the area of chip only if the tool is sharp. With a blunt tool a curve results which means a decrease in the specific chip-volume or an increase of energy as compared with these factors for sharp tools. During the test with the blunt tool the cutting pressure and the energy

N_B increased to such a degree, that the oil pressure rose to 133 kg. per sq. cm., when the safety valve blew off, although the cutting speed was only 60 m. per min. and the area of chip 3.6 sq. mm. By diminishing the area of chip without a corresponding increase of the cutting speed it was possible to continue the cutting tests until the blunted tool became entirely dull.

The highest speed of the mechanical lathe was 480 r.p.m., therefore it was not possible to get as high a cutting speed as on the hydraulic machine, and 19 sq. mm. was the largest area of chip.

From Figs. 10 and 11 the following formulas were derived for the uppermost line in each diagram:

$$\sigma_{NB} = 19.5\ F^{0.197} \text{ (hydraulic drive)} \quad (8)$$

$$\sigma_{NB} = 21.2\ F^{0.197} \text{ (mechanical drive)} \quad (9)$$

The ratio of the specific chip-volume capacities for the same area of chip is 1 to 1.09; this means that the mechanically driven lathe has a little larger specific chip-volume than the hydraulically driven one. It must be realized, however, that the hydraulic machine has a stepless speed range, while the mechanical lathe is limited by a geometric progression factor 1.26, with a 26 per cent increment between successive speeds.

Fig. 12 shows the stepless speed variations obtained with the hydraulic lathe, and the perpendicular lines represent the 18 speeds for the mechanical lathe. The oblique lines in Fig. 12 represent the permissible chip cross sections F for full load of the machines when cutting steel of 50 to 60 kg. per sq. mm. tensile strength with cutting speeds represented by the intersection of the N_B and d lines. Values are read from the oblique F-lines, as every F-line is also a v-line. The curves between the revolutions 175 and 75, 22 and 9.4 are only valid for the hydraulic machine, and represent the areas of chip in the case of pump-control. The dotted lines between these limits are valid for the chip cross-sections cut by the mechanical lathe.

Example: To make a comparison of chip cross-sections and permissible speeds with both machines cutting steel 50 to 60 kg. of tensile strength of 225 mm. dia. draw a horizontal line W-W through 225 mm. dia. Following this line from the left to the right, the oblique chip cross-section lines are interesected. Considering the hydraulic lathe one finds that for every intersection there must be a corresponding perpendicular revolution line (not drawn) while for the mechanical lathe only 18 perpendicular speed lines can be intersected, and the next lower number of revolutions must therefore be chosen if speed-line is not located exactly on the intersection point of the area of chip and the diameter. Choosing a higher speed would mean under certain conditions an overload of the machine of 30 per cent.

Comparisons for the case of full-load are continued diagrammatically. Fig. 13 shows a comparison of the possible number of revolutions, Fig. 14 shows a comparison of the obtainable chip-volumes, and Fig. 15 shows a comparison of the necessary energy. Figs. 16 and 17 compare the chip-volumes and the energy respectively for half-load. The stepless speed range of the hydraulic machine permits choosing a speed which will allow the full utilization of the machine, excepting a few points that can be found from the diagram. The same speed can be used on both machines if the curves in Fig. 13 touch one another. Therefore the chip-volume capacity of the machines is only equal for the points of contact of the curves in Fig. 14 and 16. At all other points the chip volume of the mechanical lathe is less than that of the hydraulic machine. The largest difference in the chip-volume occurs when the difference in number of revolutions is almost as large as the maximum progression of revolutions of the mechanical lathe.

At full-load the hydraulic lathe works under the most favorable conditions because of its stepless speed variation, it has a chip volume capacity 30 per cent more than that of the mechanical machine, and requires 38 per cent more energy.

In a rare case where for full-load the required speed for full utilization can also be had on the mechanical lathe, it may be cheaper to operate because less energy is required. As the most favorable condition occurs only occasionally, the best basis of comparison is a difference of from 10 to 20 per cent.

The hydraulic drive yields a 30 per cent larger chip-volume at half-load than the mechanical one and the energy required is 42 per cent higher. With both machines at half-load running at the same speed chip-volumes are equal. To the left of point E_2 in Fig. 17 the energy required increases, while the chip-volume of the hydraulic lathe increases as compared with that of the mechanical drive thus giving the hydraulic drive all the advantage.

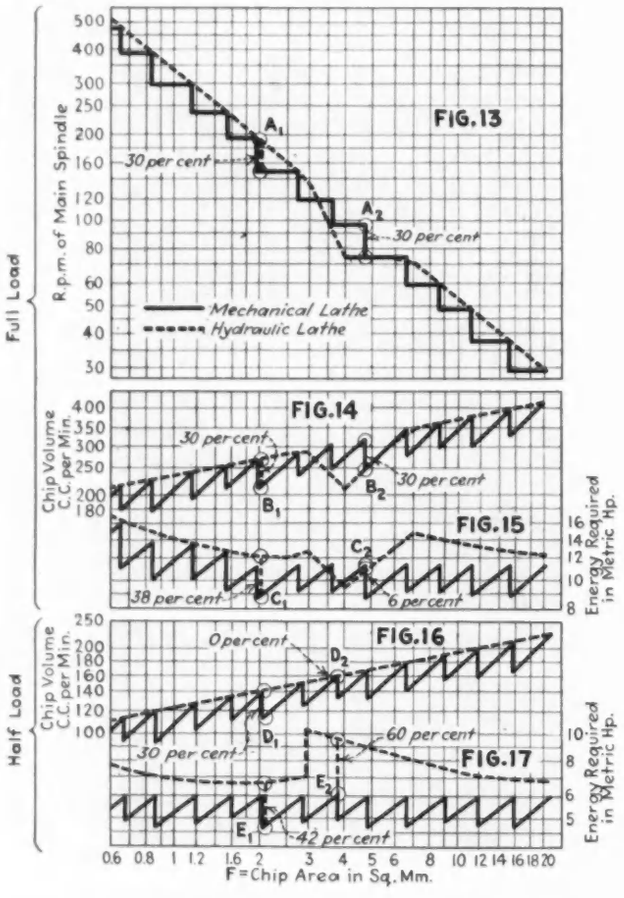
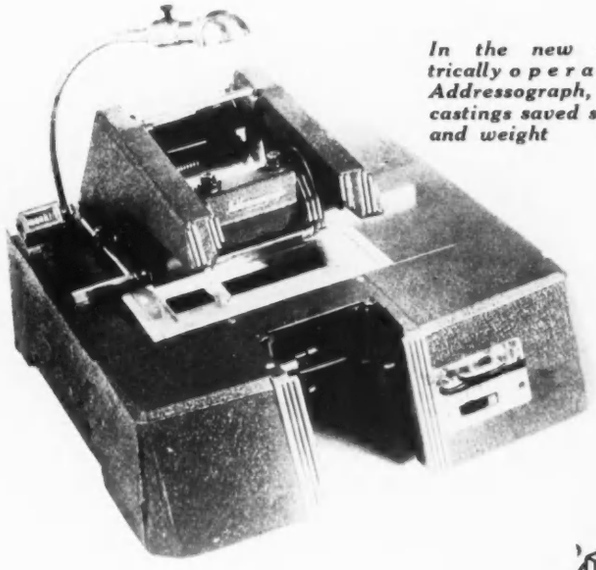


Fig. 13—Comparison at full-load, 9.6 metric hp., of machines for the permissible number of revolutions as a function of the chip area. Fig. 14—Comparison of machines for attainable chip-volume at full-load. Fig. 15—Comparison of machines for energy required at full-load. Fig. 16—Same as Fig. 14 except for half-load. Fig. 17—Same as Fig. 15 except for half-load.

PRODUCT DEVELOPMENTS

Designing for Compactness



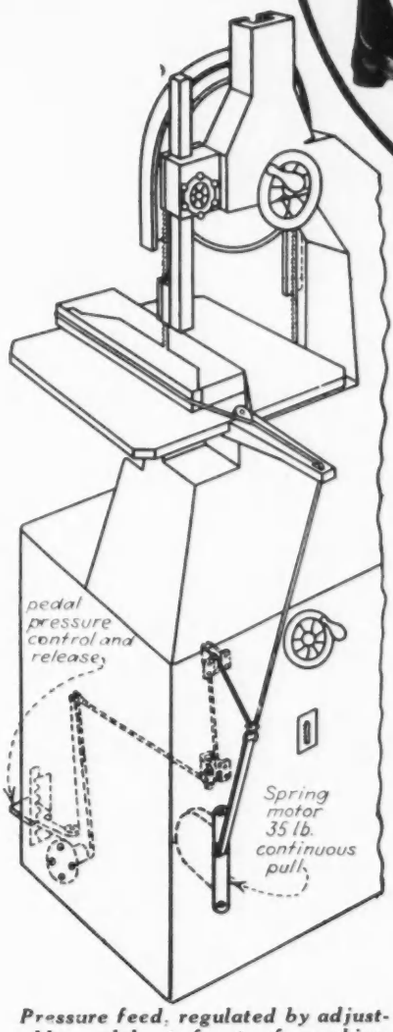
In the new electrically operated Addressograph, die-castings saved space and weight

● Because of smaller parts and closer clearances allowable with the use of zinc alloy die-castings, it was possible to design the new Model 900 Addressograph machine to occupy the same space as an ordinary typewriter. Electrically operated, the total weight of the machine is but 10 lb. heavier than a comparable manually controlled machine. Control buttons are mounted at the front of the cast-iron frame, the appearance of which is in keeping with modernistic design. Because the operating mechanism is of light weight, thereby reducing inertia forces, a speed of 1,200 to 1,500 impressions per hour is possible. A built-in light and counter facilitate operation.

● *Adapting Pressure-Feed to a Bandsaw*

To control the pressure feed when using narrow blade saws for cutting metals, the Continental Machine Specialties, Inc., have applied a power work-feed to their "Do-All" bandsaw. The power or pressure for feeding the work is furnished by a spring motor or heavy spiral spring unit which is built into the base of the machine. This spring motor applies a constant feed pressure of 35 lb., which can be controlled by a foot lever to permit further regulation of cutting pressure and instantaneous pressure release. Using a foot lever feed control allows both hands to remain free to guide the work.

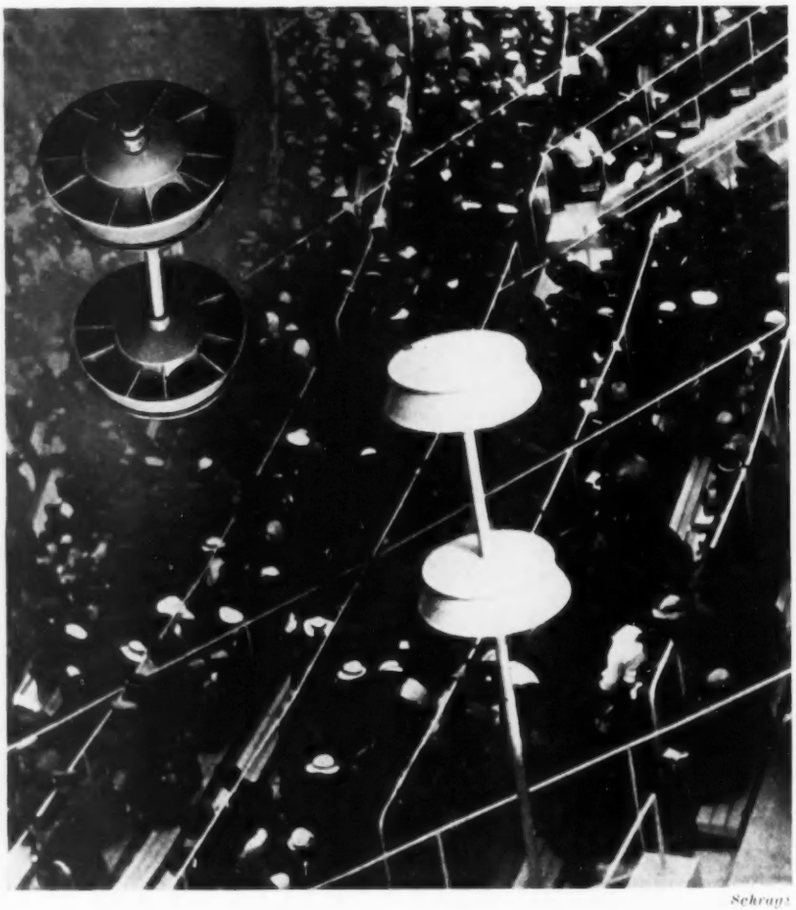
As shown in the accompanying line drawing, a stranded wire rope running from this foot lever over the small sheave on the table is connected to a welded chain noose which holds the work being sawed. A vise-like jaw can also be attached in the noose for holding any size piece of metal and permits cutting as much as 30 deg. from the center line.



Pressure feed, regulated by adjustable pedal at front of machine, controls tension in spring motor. Jaw inserted in work chain noose can hold various shapes

● *Styling Thermometers for Better Appearance*

Because of lack of harmony with decorative surroundings, it was found that thermometers are usually placed in concealed or remote posi-



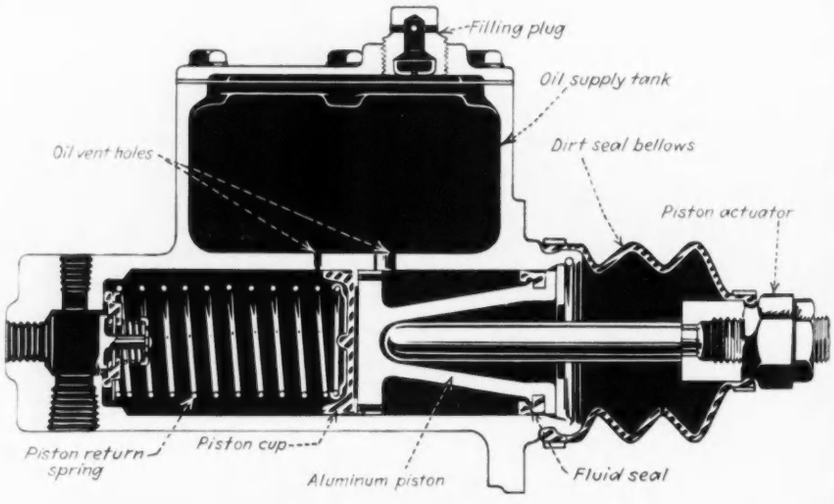
Schrage

Overlapping sounds and echoes may blur or scramble the words coming from loud speakers. Those used at the Olympic games in Berlin are tandem units such as shown here, with 5 loud speakers in each "umbrella." Insert shows how partitions, serving also as deflectors, separate the loud speakers

—50 deg. F. to +130 deg. F. The style and shape of the scale, pointers and numbers was chosen to make the expanded scale easy to read. Because of the possible exposure to adverse weather conditions, non-tarnishing satin-finished metal is used.

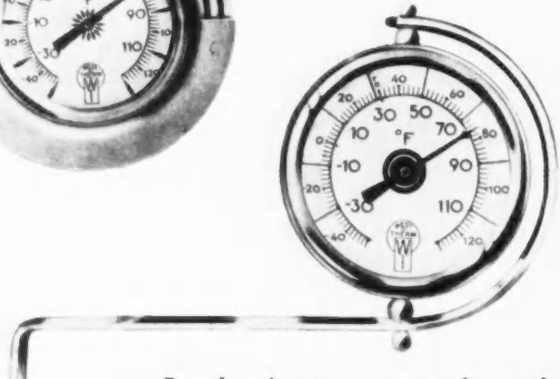
• *Aluminum Piston Forged for Strength*

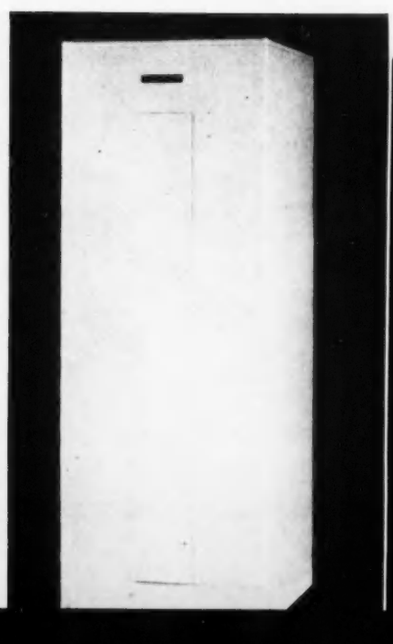
Chrysler engineers have selected a forged aluminum alloy piston for the master cylinder of their hydraulic braking system. Forging gives a dense piston that has the required strength, has a smooth surface and requires no machining. Its light weight is also an advantage.

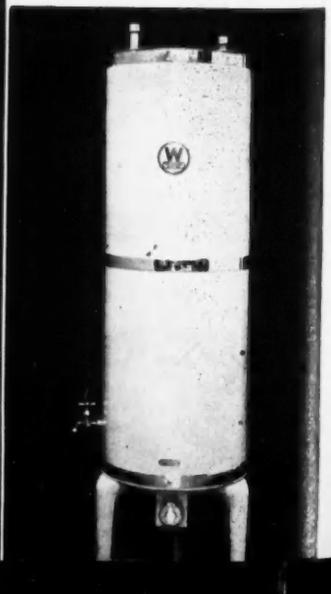


tions. To overcome this handicap, industrial styling was employed by the Weston Electrical Instrument Corporation for designs of pleasing external appearance to conform with the modern and period type of clock. These bi-metal, multi-coil thermometers were made with circular scales of the dial-and-pointer type, wherein the scale is spread almost over the full circle, being graduated from

Circular-scale thermometers styled for appearance and utility, non-corroding metal is used







By changing to a square form, the appearance of the new Westinghouse hot-water heater is improved to conform to that of the modern kitchen or basement recreation room

• *Redesigning for Improved Appearance*

Discarding the idea that a round water tank should take a cylindrical form externally, the new Westinghouse Empire line of electric hot water heaters have a square exterior. The three legs in the former design have been replaced by the black japanned square base while the upper portion is enameled white. In keeping with the remainder of the design, a single full length fluted panel conceals the two heater openings and all piping and wiring connections made within the housing.

The conventional cylindrically shaped tank is used inside the square housing, being insulated with shredded bark of the California redwood tree. Besides having a high insulating value, this bark is also rodent and vermin-proof and noncombustible. The thermostatically controlled heating elements of the direct immersion type are bolted to the adapters welded to the side of the tank.

• *Molded Rubber Ball Bearing Mounting*

Where noise elimination and economy are of utmost importance, and where loads are light, as in household heater fans, ball bearings can be mounted in a one-piece molded rubber pillow block. The housing shown here is of conventional design for a standard, steel, radial ball bearing prelubricated and shielded on both sides.

To retain the bearing after it has been pressed into the housing, a shielding plate, rubber insulating washer and thrust collar are inserted. For protection against possible oil contact, the rubber housing is lacquered. Because loads carried are small, the bearing is assembled with a light press fit and no shoulders or adapters required. It is claimed that the cost of such a rubber pedestal is less than that of an equivalent one made of cast iron.

• *Molded Plastics for Edge Protection*

Competitive markets, avoidance of welding, and pleasing appearance are factors that lead to the use of molded plastic rod ends, leg bases, table corners, and casters on Daystom tables, racks and stands. The molded black fittings are assembled with the chromium tubing by means of screw fastenings and the rod ends are shouldered to fit into the tube, the outside diameter being the same as that of the tubing. Easily assembled corners hold both the table tray and plated trim together, thereby avoiding all sharp metal edges and preventing scratching.

To exclude extraneous sounds sponge rubber is wrapped about the newly designed German crystal microphone having a preamplifier in the tube. The microphone is used in the motor launch following the crews at the Olympic regatta

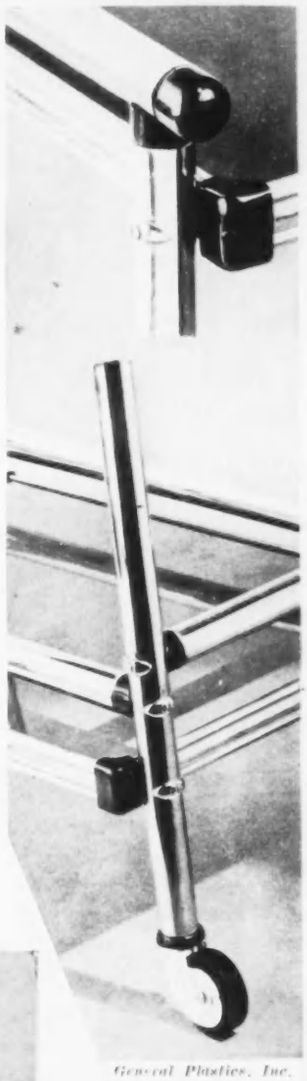
Schrage

• *Random Jottings About New Developments*

By changing the material from high carbon, high alloy tool steel, to nickel nitrided steel, 57 per cent greater production could be obtained from a set of dies used in an automatic roll threader. The nitrided steel dies contained 0.20 carbon, 3.5 nickel, 1.5 chromium, 0.25 molybdenum and 1.00 aluminum. The material was first heat-treated to a maximum machineable hardness and then nitrided for 48 hr. at 975 deg. F. and lapped.

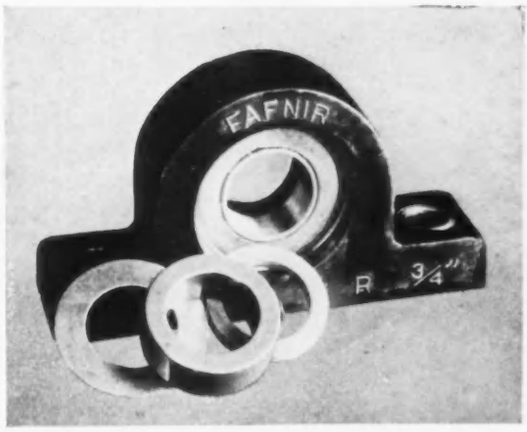
* * *

It is predicted that greater use will be made of stainless steel and molded plastics in 1937 automobiles. The former will be used for body moldings and other parts, the latter replacing steel particularly for window trim. Die-cast radiator grilles will also be retained.



General Plastics, Inc.

Molded plastic rod ends and tray corners prevent sharp edges and simplify the assembly of the chromium plated tubes



Molded rubber ball-bearing pillow block, lacquered to make it oil resistant, can be used for light loads. Bearing pre-lubricated and sealed.

How to Make Alignment Charts

M. G. VAN VOORHIS

Concluding article of the series with further data for laying out nomograms with curved scales, also showing the use of a "curve net" or family of curves for chart construction

PART VII

IN THE construction of curved scale nomograms an infinite variety of shapes can be produced in the curved scale line by shifting the position of one scale. In Fig. 48 is shown a nomogram having four h scales, each h scale being plotted by shifting the position of the working scale N (not shown). The nomogram represents the equation:

$$V = \frac{2}{3}\sqrt{2g}\,\frac{\sqrt{h_1^3} - \sqrt{h_2^3}}{h_1 - h_2}$$

These four curves are drawn between the V and N scales, two values of h on the curved scale determining a value of V on the V scale.

The nomogram for determining the volume of a frustrum of a pyramid, useful in bin and hopper design, as shown in Fig. 49, is of interest, as four variables are involved. The principle is similar to that employed for constructing the nomogram shown in Fig. 48. The formula which the chart solves is

$$V = \frac{h}{3}(A_1 + A_2 + \sqrt{A_1 A_2})$$

The variables A_1 and A_2, being the areas of the top and bottom of the pyramid, have identical scales superimposed upon each other. The distance between the top and bottom, h, is scaled on the diagonal line and the volume V of the pyramid is indicated on one of the vertical scale lines.

As an example of the use of the chart a top area of one unit and a bottom area of 6 units are assumed. A line drawn through these two points on the A_1 and A_2 scale line intersects the vertical line at the right. A second line drawn from the determined point through the value $h = 4$ on the diagonal scale yields a value of 12.6 units for volume.

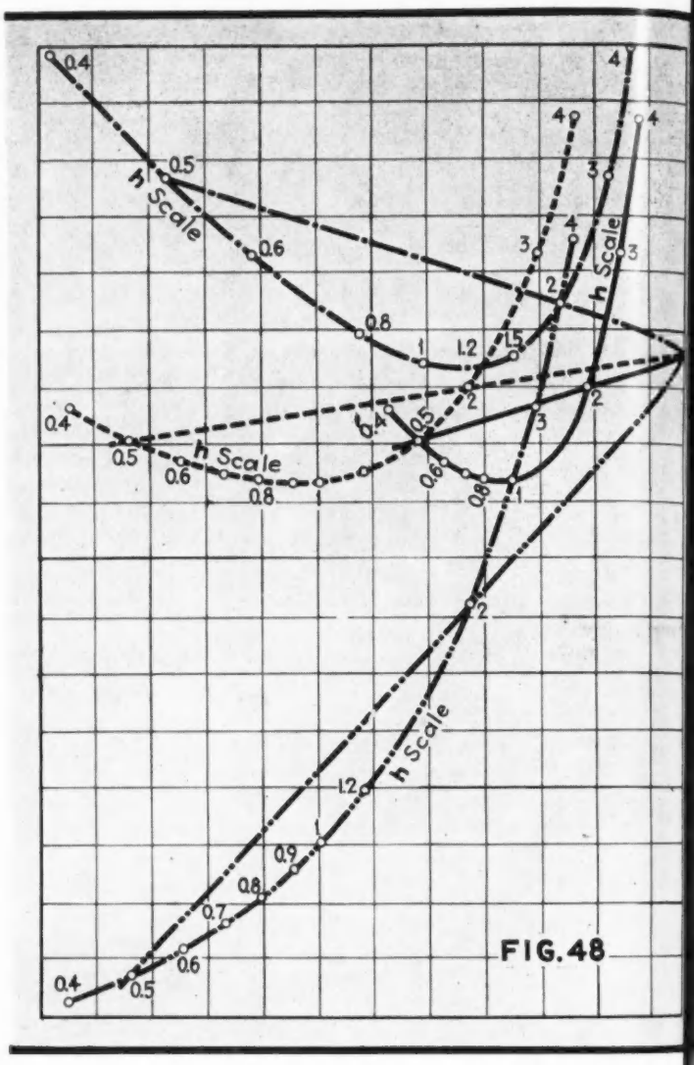


FIG. 48

The formula can be changed to the form

$$3V/h = (\sqrt{A_3} - \sqrt{A_3'})/(\sqrt{A_1} - \sqrt{A_2})$$

by multiplying the equation for volume by

$$(\sqrt{A_1} - \sqrt{A_2})/(\sqrt{A_1} - \sqrt{A_2})$$

This equation can also be written:

$$\frac{3V}{h}\sqrt{A_1} - \sqrt{A_1^3} = \frac{3V}{h}\sqrt{A_2} - \sqrt{A_2^3}$$

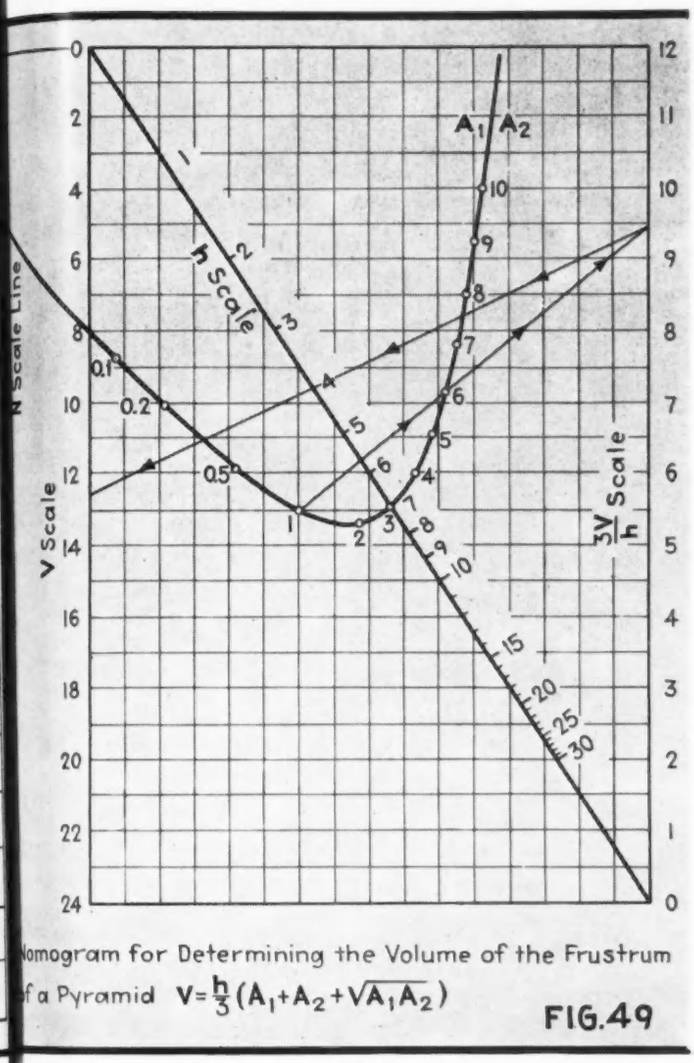
Using the variable A for A_1 or A_2 and introducing N

$$\frac{3V}{h}\sqrt{A} - \sqrt{A^3} = N$$

A nomographic type form is derived by dividing through the equation by $\sqrt{A^3}$

$$\left(\frac{3V}{h}\right)\left(\frac{1}{A}\right) + N\left(\frac{-1}{\sqrt{A^3}}\right) = 1$$

In the accompanying chart, limits of 0 to 12 were selected for the ratio $3V/h$ which is handled as a single variable. It is a matter of trial and error to determine suitable limits of N and the position of the N scale line. As indicated in Fig. 48, the possibilities are without limit. A portion of the N scale line is shown with the chart but it has no function in the completed



Nomogram for Determining the Volume of the Frustrum of a Pyramid $V=\frac{h}{3}(A_1+A_2+\sqrt{A_1A_2})$

FIG. 49

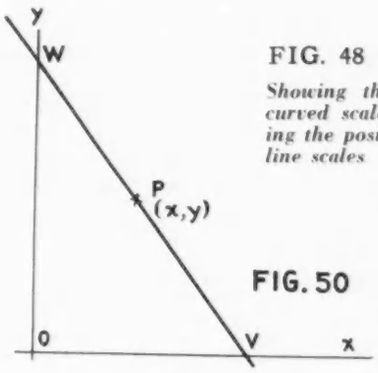


FIG. 50

FIG. 48

Showing the various shapes of the curved scale line developed by shifting the position of one of the straight line scales

FIG. 49

Nomogram used in hopper or bin design for determining the volume of the frustrum of the pyramid formed by the walls

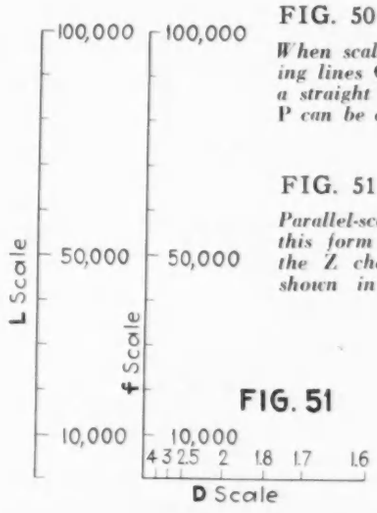


FIG. 51

FIG. 50

When scales are placed on intersecting lines Oy and Ox, the equation of a straight line passing through point P can be developed

FIG. 51

Parallel-scale type of nomogram in this form is more complicated than the Z chart for the same formula, shown in Fig. 12, March number

FIG. 52

Nomogram of equation $w=u\sqrt{u^2+1}\sqrt{v^2+1}$ *after transformation to workable type form*

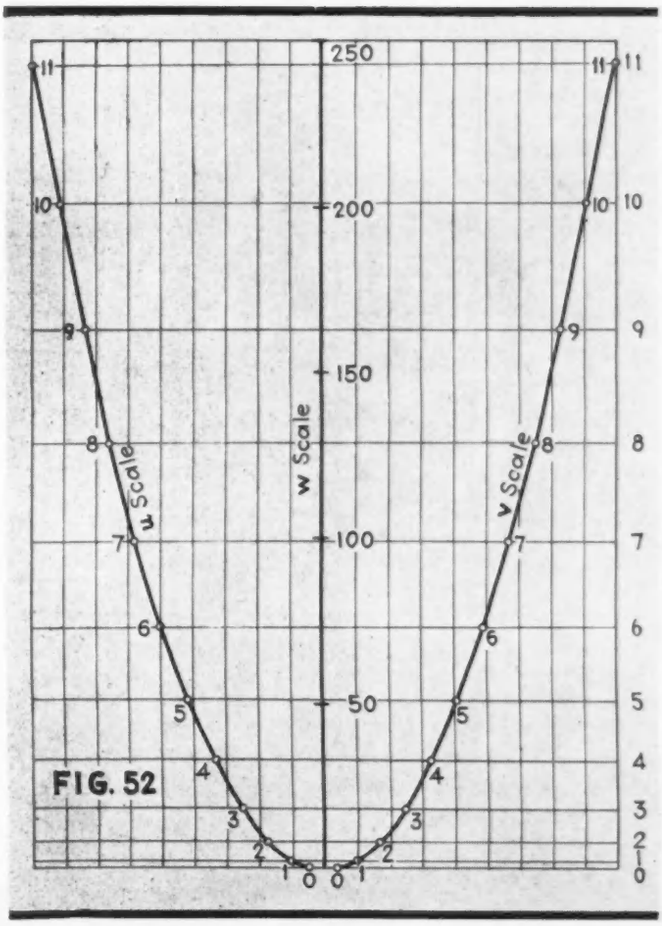


FIG. 52

chart and may be eliminated with the numbers on the ratio scale line.

The $3V/h$ scale line serves as the y axis and a line drawn horizontally through zero on this line is the x axis. The N scale line is at $x = -1$ and the N scale ascends in a direction opposite to the $3V/h$ scale. The N scale has also been raised 10 units to bring it opposite the other vertical scale. The x and y coordinates of the A_1 and A_2 scale were determined from the formulas:

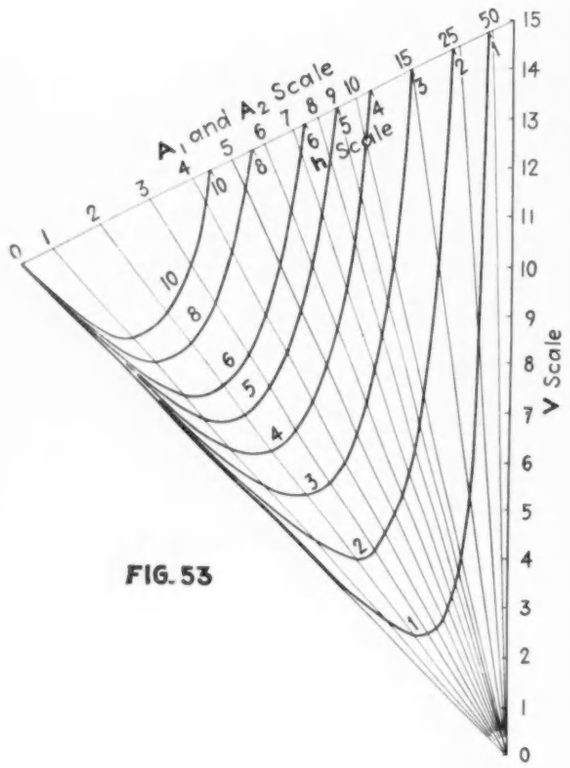
$$x = \frac{-1}{\sqrt{A}+1} \text{ and } y = \frac{\sqrt{A^3}}{\sqrt{A}+1}$$

These apply before raising the N scale 10 units. If the N scale is raised 10 units, y' is substituted for y ($y' = y - 10x$) while x remains the same.

$$y' = \frac{\sqrt{A^3}+10}{\sqrt{A}+1}$$

A	x	y'	A	x	y'
0	−1	10.00	4	−0.333	6.00
0.1	−0.760	7.62	5	−0.309	6.55
0.2	−0.691	6.97	6	−0.290	7.16
0.5	−0.586	6.07	7	−0.274	7.81
1	−0.500	5.50	8	−0.261	8.52
2	−0.414	5.31	9	−0.250	9.25
3	−0.366	5.56	10	−0.240	10.00

The A_1 and A_2 scale was plotted from the accompanying data. Computed in this manner, it is advan-



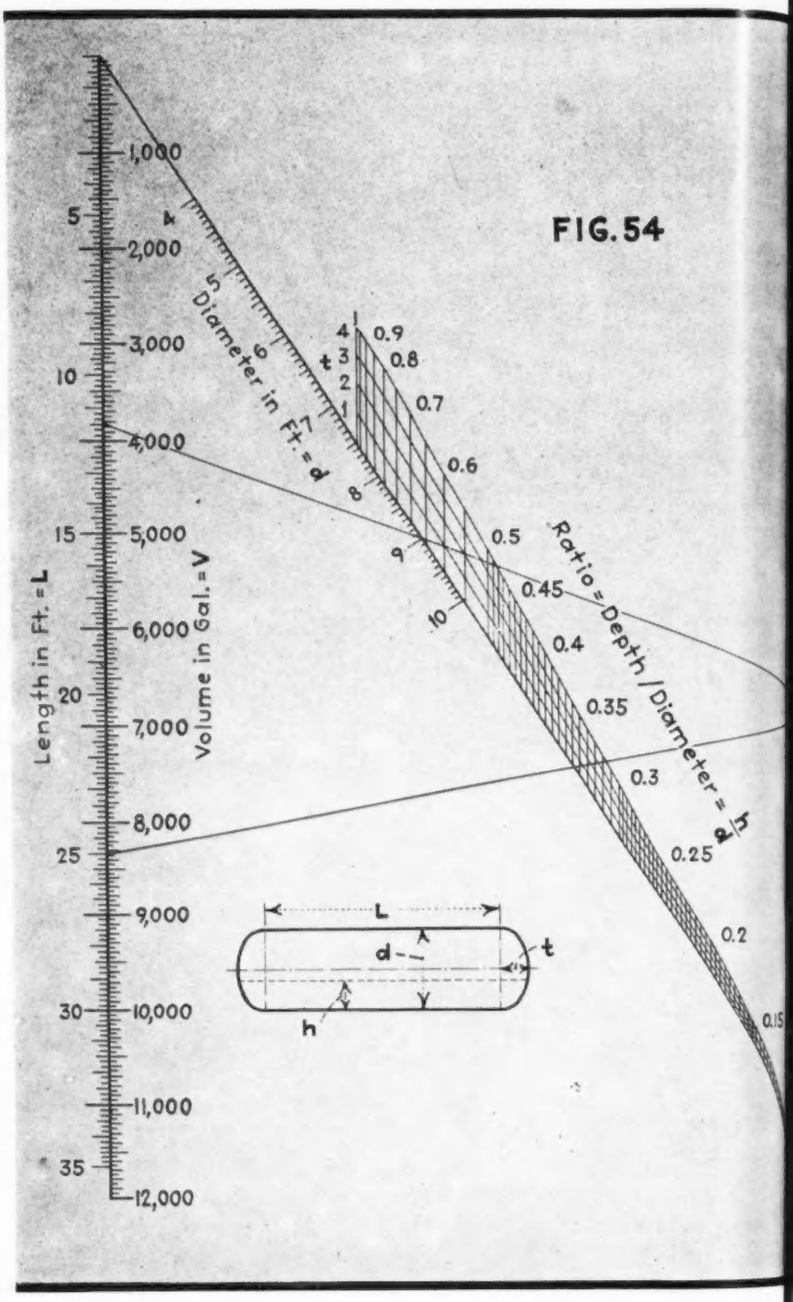


FIG. 53

Showing the use of the "curve net" or family of curves in chart construction

FIG. 54

Chart for determining volume of liquid at any depth in a horizontal cylindrical tank with spheroidal ends

FIG. 55

Nomogram used for predicting the viscosity of blends of oil of the same base when high and low viscosity oils are mixed

tageous to lay out the chart on cross-sectioned paper.

The second half of the equation, involving V and h, is simply the Z chart. It cannot be a logarithmic scale chart because the $3V/h$ scale is uniformly divided for the other part of the chart.

When scales are placed on intersecting lines such as the abscissa and ordinate in Fig. 50, the general equation for a straight line through any point, P, may be written $(y_1—y)/(x_1—x) = (y—y_2)/(x—x_2)$. The point (x_1, y_1) is at w and the point (x_2, y_2) is at v. The relation is obvious from the theorem of similar triangles. Then, $(w—y)/(O—x) = (y—O)/(x—v)$, and by clearing fractions, dividing by vw; and rearranging, the equation becomes $(x/v) + (y/w) = 1$.

In this equation, v and w represent any points on the x and y axes and x and y the co-ordinates of points on a curved scale of u. An equation of the form $f_1(u)/f_3(v) + f_2(u)/f_4(w) = 1$ can be applied directly to the construction of a nomogram on intersecting axes by letting $x = f_1(u)$, $y = f_2(u)$, and placing uniform scales of $f_3(v)$ and $f_4(w)$ on the x and y axes, respectively.

It was previously stated that methods applicable to equations in which there are two functions of one of the variables are applicable to the simpler nomograms. It is interesting to see how this works out. As an example, the equation:

$$D = 1.24\sqrt{L f} + 0.088$$

one form of which was illustrated in Figs. 12, and 13, page 101 in the March number, will be used. First, the parallel scale type will be considered. The equation may be written:

$$\left(\frac{D - 0.088}{1.24}\right)^2 f + (-1)(L - 1) = 1$$

$$f_1(u) = \left(\frac{D - 0.088}{1.24}\right)^2; f_2(u) = -1; f_3(v) = f; \text{ and}$$

$$f_4(w) = L - 1$$

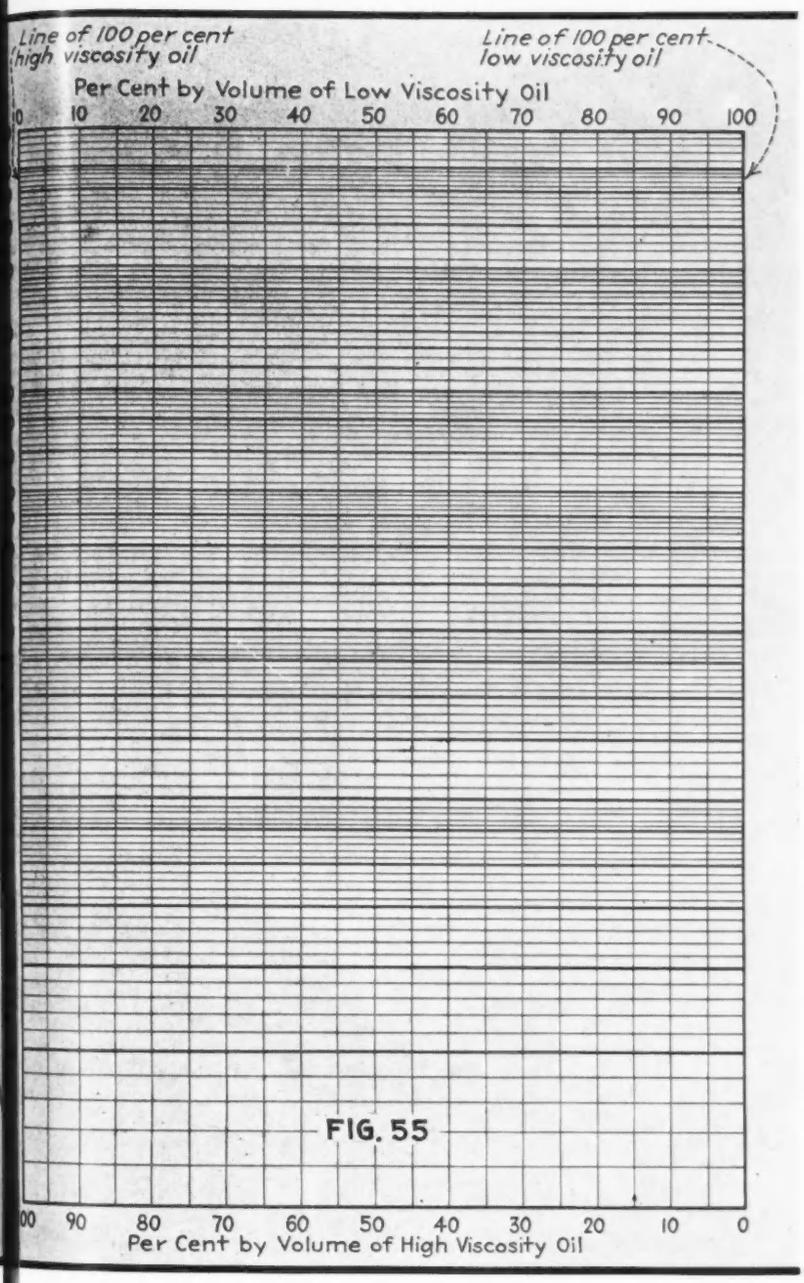

Line of 100 per cent high viscosity oil
Line of 100 per cent low viscosity oil
Per Cent by Volume of Low Viscosity Oil
0 10 20 30 40 50 60 70 80 90 100
100 90 80 70 60 50 40 30 20 10 0
Per Cent by Volume of High Viscosity Oil

FIG. 55

By referring to the plotting method on page 270 of the July issue, it will be seen that:

$$x = \frac{1}{[(D - 0.088)/1.24]^2 - 1}$$

$$= \frac{1.538}{D^2 - 0.176D - 1.53} = y$$

The resulting nomogram is shown in Fig. 51. Although x and y values are numerically the same, the D scale is practically horizontal because of the great difference in size of the x and y units. When $D = 1.6$ for example, $x = y = 2.06$. If

$$f_2(u) = 1 \text{ and } f_1(w) = -(L-1)$$

$$= \frac{-1}{[(D - 0.088)/1.24]^2 + 1} = \frac{-1.538}{D^2 - 0.176D + 1.545}$$

$$y = \frac{1}{[(D - 0.088)/1.24]^2 + 1} = \frac{1.538}{D^2 - 0.176D + 1.545} = -x$$

If the L-scale is then raised 99,999 units, then $y' = 100{,}000\ y$ and the resulting chart will be as shown in Fig. 13, page 101 in the March issue. The equation could also be written,

$$2 \log [(D - 0.088)/1.24] = \log L - \log f$$

$$\text{or } (\log L)/f_1(D) + (-\log f)/f_1(D) = 1.$$

$$x = \frac{-1/f_1(D)}{1/f_1(D) + 1/f_1(D)} = -\frac{1}{2}$$

$$\text{and } y = \log\left(\frac{D - 0.088}{1.24}\right)$$

Obviously, the resulting nomogram of the last equation would consist of equal sized logarithmic scales of L and f ascending in opposite directions on parallel lines with the D scale, as defined by the equation for y above, on a third parallel line half way between them. Similarly the equation could be arranged for intersecting scales of L and f by writing it in the form, $f_1(u)/f_3(v) + f_2(u)/f_4(w) = 1$. Then $f_1(u) = f_2(u) = 1/f_1(D)$.

$$x = 1/f_1(D) \qquad f_3(v) = 1/\log L$$

$$y = 1/f_1(D) \qquad f_4(w) = -1/\log f$$

But the resulting nomogram would be unwieldly and impractical.

Just as it has been shown possible to construct all nomograms from a theory set up for the more complex types, it is also possible to build a theory around determinants which is applicable to all nomograms. The algebraic manipulation becomes very cumbersome and voluminous by either method for the more general equations so that for practical applications the labor involved in construction is rarely justified. Transformations of equations into type forms are frequently puzzle problems if they are workable at all. It has been stated that no test has been found which could show definitely if it is possible to reduce certain equations to the most general form.

Some equations have been put into nomographic form without benefit of theory where two functions of two variables are in symmetrical relation. An example of this kind is the equation:

$$w = uv + \sqrt{u^2 + 1}\ \sqrt{v + 1}$$

It is obvious how this equation could be put into the type form of

$$f(w) = \frac{f_1(u)\ f_2(v) - f_2(u)\ f_1(v)}{f_1(u) - f_1(v)}$$

$$\text{or} \quad f(w) = \frac{f_1(u) - f_1(v)}{f_2(u) - f_2(v)}$$

Yet the nomogram in Fig. 52 solves the equation. Being constructed graphically, the first set of construction lines are drawn assuming $u = v$ and the values of u and v assumed to be on lines drawn horizontally through values of w on a uniform scale as determined by the formula, $w = 2u^2 + 1$. The second set is assumed to pass through a constant value of u or v, for example, $u = 10$, which is located as far from the w scale as desired. The equation would be:

$$w = 10u + \sqrt{101}\ \sqrt{u^2 + 1}$$

The fault in the theory of construction is that there is no proof for any value other than 10. However, when a similar scale is made for v, the nomogram is found to be practicable for all values.

The method holds true also with the equation:

$$Z = K\,\frac{A^3\,B^3}{A+B}$$

both with a uniform scale of Z and with a logarithmic scale, although the method is known to be impracticable in other similar equations.

At this point it may be recalled that equation

$$V = \frac{2}{3}\,\sqrt{2g}\,\frac{\sqrt{h^3{}_1} - \sqrt{h^3{}_2}}{h_1 - h_2}$$

as in Fig. 48, is in the abbreviated type form of a two-curve-scale nomogram. The equation for the curves of h_1 and h_2, however, will be found to be identical and therefore lie in the same position so that the resulting nomogram would be the same as that in Fig. 48. If the signs between h_1 and h_2 had been plus, two curves would have been necessary.

When more than three variables are present in an equation they may be handled as in Fig. 49, for example, or by using a "curve net" or family of curves. The equation for the volume of a frustrum of a pyramid would appear as in Fig. 53 when a "curve net" is used. In this instance the result is not satisfactory because the range of h is limited for reasonable accuracy and because values of A less than unity are so close together that they are indistinguishable.

A nomogram of this type which is satisfactory is shown in Fig. 54. There are five variables in the equation which is represented. It gives not only volumes of liquid in cylindrical tanks for any depth of liquid but, with a simple side calculation it gives the volumes in tanks of elliptical cross-section. Flat-end tanks are also included.

The equation is:

$$\frac{4V}{d^2} = \frac{1728}{231}\,4\pi t\left(\frac{h}{d}\right)^2\left(1 - \frac{2h}{3d}\right) + L\cos^{-1}\left(1 - \frac{2h}{d}\right) - 2\left(1 - \frac{2h}{d}\right)\sqrt{\frac{h}{d} - \left(\frac{h}{2}\right)^2}$$

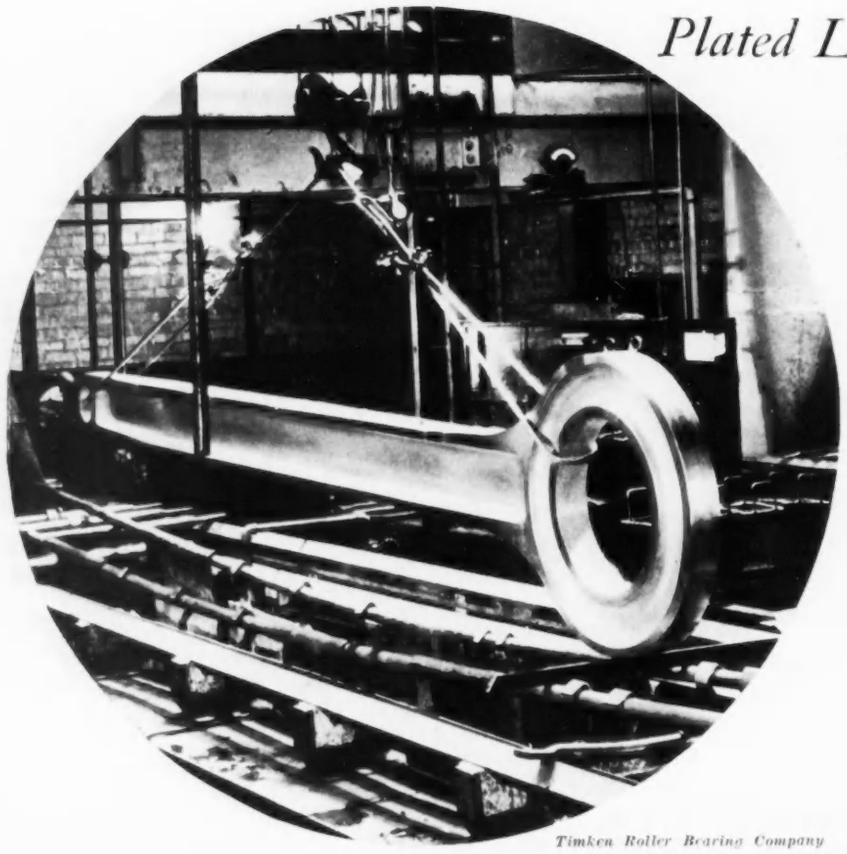
V = volume in gal.
d = diameter in ft.
L = length in ft.
h = depth of liquid in ft.
t = additional length on the center line of the tank because of the spheroidal ends.

A "curve net" nomogram, which does not look like a nomogram at all, is shown in Fig. 55, but it solves the equation:

$$\frac{100}{6.5\sqrt{V}} = \frac{P}{6.5\sqrt{v_1}} + \frac{(100 - P)}{6.5\sqrt{v_2}}$$

The formula gives the kinematic viscosity, V, of a blend of two oils from the same base of viscosity v_1 and v_2 at the same temperature of the blend when P is the percentage by volume of the higher viscosity component. Fundamentally it is an equation of the type, $2f(V) = f(v_1) + f(v_2)$, when $P = 50$. Normally the $f(V)$ scale would be half the size of the $f(v_1) + f(v_2)$ scales but the figure 2 in the equation doubles its size. All functions being identical, the scales also are identical. Similar reasoning could be applied if the values of P were other than specified.

This concludes the series on "How to Make Alignment Charts." The preceding installments have appeared this year in the January, February, March, May, July and August numbers.



Timken Roller Bearing Company

Plated Locomotive Main Rods for Flood Lighting

TO EMPHASIZE the speed of the new high-speed train "Mercury," of the New York Central System, the side and main rods are chrome plated and illuminated by hidden floodlights. Said to be the largest pieces ever chrome plated in a single operation, a specially designed plating tank 3 ft. x 3 ft. 17 ft. had to be constructed. The plating current is 6,500 amp. at 6 volts and provision is made to control the temperature of the work and the bath to assure a uniform and dense coating of chromium. To carry away the fumes, a 7½ hp. exhaust system having a capacity of 12,000 cu.ft. of air per min. was installed, provision being made to salvage the chromic acid from the fumes that pass through the system by a washing process.

Analysis and Comparison of Cam Actions

R. E. GROBE
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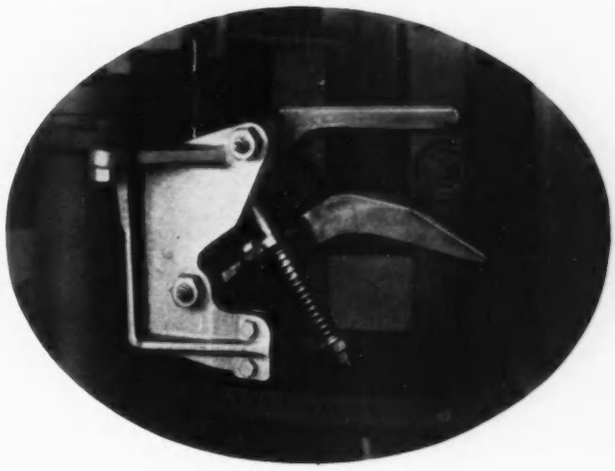
Three basic profiles are graphically investigated for inherent advantages and for points of shock by analysis of displacement, velocity, and acceleration curves



WHEN selecting a cam profile to use to give intermittent or reciprocating motion to machine parts, the points of shock can be determined from a study of the displacement, velocity and acceleration curves of the profiles under consideration. A slight change in the profile will account for a considerable difference in the action. A comparison of the common harmonic, parabolic and "t-sin t" displacement motions will illustrate the method, and present graphically the characteristics inherent in each for any particular application.

The total displacement S is taken as 2 units for each of the three curves. Time t is measured from the beginning of stroke in degrees or radians and is taken to be 360 deg. or 2π radians as a linear measurement from one position of rest to the next position of rest of the cam-follower. Time forms the base for the curves, it is entirely independent of the angle of rotation of the camshaft that produced motion of the follower.

Displacement or stroke s of the cam-follower is given for any position t (expressed in radians) as

harmonic displacement $s_h = 1 - \cos(t/2)$, (1)

parabolic displacement $s_p = (t/\pi)^2$ (2)

and "t — sin t" displacement $s_s = (t - \sin t)/\pi$ (3)

When t is expressed in degrees of cycle, Equations (1), (2) and (3) become

harmonic displacement $s_h = 1 - \cos(t/2)$ (4)

parabolic displacement $s_p = (t/180)^2$ (5)

and "t — sin t" displacement $s_s = [(t/180) - \sin t]/180$ (6)

These formulas are valid only for the period of acceleration up to the point of inflection, or half the total displacement S, since the motions of acceleration and deceleration are symmetrical about this point. The portions of the curves shown in Fig. 1 in full lines are for acceleration. The dotted line portions show deceleration, they are obtained by inverting and reversing the full line portions.

By differentiating the s equations for displacement, the velocity curves shown in Fig. 2 are obtained for t expressed in radians, or

harmonic velocity $v_h = \frac{ds_h}{dt} = (1/2)\sin(t/2)$ (7)

parabolic velocity $v_p = \frac{ds_p}{dt} = (2t)/\pi^2 = 0.202\,t$ (8)

"t — sin t" velocity $v_s = \frac{ds_s}{dt} = (1 - \cos t)/\pi$ (9)

When t is expressed in degrees of cycle, Equations (7), (8), and (9) become

harmonic velocity $v_h = \frac{ds_h}{dt} = (1/2)\sin(t/2)$ (10)

parabolic velocity $v_p = \frac{ds_p}{dt} = (2t)/180^2 = t/16{,}200$ (11)

"t — sin t" velocity $v_s = \frac{ds_s}{dt} = [(1/180) - \cos t]/180$ (12)

By differentiating the v equations for velocity, the acceleration curves shown in Fig. 3 are obtained, for t expressed in radians

harmonic acceleration $a_h = \frac{dv_h}{dt} = (1/4)\cos(t/2)$, (13)

parabolic acceleration $a_p = \frac{dv_p}{dt} = 0.202$ (14)

"t — sin t" acceleration $a_s = \frac{dv_s}{dt} = (\sin t)/\pi$ (15)

When t is expressed in degrees of cycle, Equations (13), (14), and (15) become

harmonic acceleration $a_h = \frac{dv_h}{dt} = (1/4)\cos(t/2)$ (16)

parabolic acceleration $a_p = \frac{dv_p}{dt} = 1/16{,}200$ (17)

"t — sin t" acceleration $a_s = \frac{dv_s}{dt} = (\sin t)/180$ (18)

The maximum velocities produced by the parabolic and "t — sin t" motions are equal and slightly greater than

Value of Ordinates for "t-sin t" Displacement Curve

Number of Ordinates in Curve to Inflection Point										Value of $(t-\sin t)/\pi$
5	6	8	9	10	12	15	16	18	20	
0	0	0	0	0	0	0	0	0	0	0.00000
									1	0.00020
								1		0.00029
							1			0.00041
						1				0.00047
					1					0.00095
				1					2	0.00165
			1					2		0.00226
		1					2			0.00318
						2				0.00388
									3	0.00546
	1				2			3		0.0075
							3			0.0098
1				2		3			4	0.01289
			2					4		0.0176
		2			3		4		5	0.0249
						4				0.0301
								5		0.0339
				3					6	0.0425
							5			0.0478
	2		3		4	5		6		0.0576
									7	0.0664
		3					6			0.0810
								7		0.0896
2				4		6			8	0.0971
					5					0.109
							7			0.125
			4					8		0.131
									9	0.1355
						7				0.1500
	3	4		5	6		8	9	10	0.182
						8				0.2160
									11	0.235
			5					10		0.242
							9			0.250
					7					0.276
3				6		9			12	0.297
								11		0.312
		5					10			0.330
									13	0.366
	4		6		8	10		12		0.390
							11			0.423
				7					14	0.441
								13		0.477
						11				0.495
		6			9		12		15	0.524
			7					14		0.574
4				8		12			16	0.612
							13			0.634
	5				10			15		0.673
									17	0.705
						13				0.735
		7					14			0.752
			8					16		0.779
				9					18	0.800
					11					0.833
						14				0.866
							15			0.873
								17		0.885
									19	0.900
5	6	8	9	10	12	15	16	18	20	1.00

that by harmonic motion. This fact is not of great practical importance since any undesirable conditions resulting from the velocity of the parts according to one motion, could hardly be expected to be rectified by using either of the others as an alternative.

Because of the change in the magnitude of the force required to change the motion of a mass, objectionable shocks may result from any sudden changes in acceleration or deceleration, since the force necessary to move a body or to bring it to rest, is directly proportional to the product of the mass and the acceleration.

In harmonic motion, as shown in Fig. 3, there are two points of shock since the acceleration at the beginning of the rise increases abruptly to a maximum, then decreases gradually to zero at the inflection point to a maximum negative value at the top of rise, and then decreases suddenly to zero. In parabolic motion there are three points of shock, here also the acceleration at the beginning of the rise increases directly from zero to a value which is constant up to the point of inflection where it precipitately reverses to a constant negative value which is maintained until the top of rise where it immediately reverses to zero. Note the different action in the "t — sin t" curve. Here there is no sudden increment or decrement of acceleration, the curve is smooth gradually changing throughout from zero at the start to a maximum at about quarter-stroke, then decreases to zero at the inflection point, and decelerates with a maximum value at three-fourth stroke and zero at top of rise. In any application the "t — sin t" curve will produce the smoothest action.

To facilitate the plotting of any "t — sin t" displacement curve Table I is included. The values of $(t - \sin t)/\pi$ given are ordinates for a number of evenly spaced divisions of the period or time base up to the inflection point, for a total rise of 2 units. Since the ordinates at the same division for rises of different amounts will always be proportional to the rise, the values of $(t - \sin t)/\pi$ in the table can be used for any rise simply by multiplying the tabled value by the particular rise desired and dividing by 2. To obtain the portion of the cam curve beyond the inflection point, the lower portion of the curve is inverted and reversed.

If desired with the "t — sin t" curve the inflection point can be located anywhere between the top and bottom of the rise, in which case however the two portions of the curve would not be inversely symmetrical. Each portion can be plotted using the values of $(t - \sin t)/\pi$ from Table I and proportioning these values with a factor determined by the ratio of twice the numerical value of the rise at point of inflection and the basis 2 on which the Table is calculated.

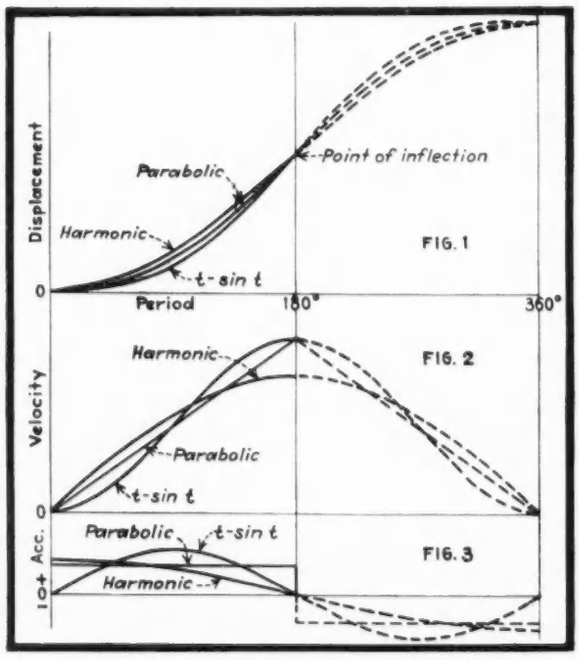


Fig. 1—Equal stroke curves plotted on one base to compare displacement at all points of the cycle. Fig. 2—Maximum velocity is at point of inflection, parabolic and "t-sin t" motions have same maximum velocity with maximum for harmonic about one-fifth slower. Fig. 3—Sudden changes of values in acceleration curves indicate point of shock

Forces and Accelerations in Slider-Crank Mechanisms

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After a brief derivation of the equations involved, the author gives tables for simplifying the calculations and shows by an example, how they are used. Although a diesel engine is used for the example, the tables and equations are applicable, with slight modifications, to any type of reciprocating engine or compressor and any straight-line slider-crank mechanism. The tables appearing on pages 323 and 324 of the August *P. E.* and on pages 363 and 364 of this number, form a part of this article

IN DETERMINING the position, velocity and acceleration of the piston or the crank moment of a reciprocating engine or equivalent mechanism, the designer frequently finds himself repeating calculations that he has previously performed on some other engine. Such calculations usually involve troublesome graphical methods or mathematical formulas that require considerable time to solve. Arithmetical mistakes are likely to occur. To simplfy the solution of such problems this acticle presents tables of factors which have been computed for an engine of unit size. By simple multiplication, the piston acceleration, crank moment and inertia torque for a reciprocating machine of any dimensions can be calculated.

The reciprocating machinery for which the tables are primarily intended include gas engines, diesel engines, steam engines, air compressors, ice machines, and others of similar construction, but are applicable to any slider-crank mechanism wherein the center line of the piston or crosshead intersects the crankshaft center line. This covers most of such mechanisms, as engines are now seldom built with an "offset" crank. However, for a machine having a moderate degree of "offset" the error resulting from the use of the following tables will be very slight.

A table of factors for determining the position of the crosshead or piston for different crank positions and a table for determining the velocity of the piston and the crank moment at different crank positions, have been published. (See p. 323 and 324, Aug. 1936 *P. E.*)

The acceleration of the piston or crosshead is the derivative of its velocity with respect to time, or

$$a = -r\left(\frac{d\theta}{dt}\right)^2 (q_1 \cos\theta + 4q_2 \cos 2\theta + 16q_4 \cos 4\theta + \ldots)$$

from which $a = r\omega^2 F_3$ 1)

where minus F_3 is the last bracketed term in the first equation.

In dealing with the crank moment it is convenient to assume that the net friction forces—the resistance at the piston, piston pin bearing, crankpin bearing, etc.—give a resisting torque which does not vary during the revolution of the engine. The torques caused by gas pressure, inertia, and gravity forces may now be determined alone, and the constant friction moment assumed to be concentrated at some point beyond the crank, say at the main bearing journals.

The forces affecting the torque of a reciprocating machine are as follows:

1. Gas pressure on the piston.
2. The force required to accelerate the mass of the reciprocating parts (piston, rings, piston pin, and upper end of the connecting rod).
3. A residual force on the crank pin which corrects for an assumption that the weight of the connecting rod is concentrated at the crank pin and piston pin.
4. The weight of the reciprocating part of the connecting rod and of the unbalanced crank throw.

In considering the gas pressure on the piston, let P be the total gas pressure acting on the piston in the line of stroke. The rate of work done on the piston by

Table V.—Calculation of Crank Moments.

A — CRANK ANGLES IN DEGREES

Cyl. No. 1	0	15	30	45	60	75	90	105	120	135	150	165	180
Cyl. No. 2	540	555	570	585	600	615	630	645	660	675	690	705	720
Cyl. No. 3	180	195	210	225	240	255	270	285	300	315	330	345	360
Cyl. No. 4	360	375	390	405	420	435	450	465	480	495	510	525	540

B — PISTON POSITIONS IN PER CENT OF COMPLETED STROKE

Cyl. No. 1	0.0	2.1	8.2	17.6	29.4	42.6	55.9	68.4	79.4	88.3	94.8	98.7	100.0
Cyl. No. 2	100.0	98.7	94.8	88.3	79.4	68.4	55.9	42.6	29.4	17.6	8.2	2.1	0.0
Cyl. No. 3	100.0	98.7	94.8	88.3	79.4	68.4	55.9	42.6	29.4	17.6	8.2	2.1	0.0
Cyl. No. 4	0.0	2.1	8.2	17.6	29.4	42.6	55.9	68.4	79.4	88.3	94.8	98.7	100.0

C — CYLINDER PRESSURES FROM INDICATOR DIAGRAM

Cyl. No. 1	457	637	432	269	171	116.5	82	64	55.5	50	44.5	41	36
Cyl. No. 2	0	0	0	0	0	0	9	21.5	42	76	151	297	457
Cyl. No. 3	36	26	14.5	8	7	0	0	0	0	0	0	0	0
Cyl. No. 4	0	0	0	0	0	0	0	0	0	0	0	0	0

D — CRANK MOMENTS FOR UNIT SIZE ENGINE CYLINDERS

Cyl. No. 1	0.0	202.0	259.8	222.0	165.7	119.5	82.0	58.0	42.4	29.5	17.7	8.2	0.0
Cyl. No. 2	0.0	0.0	0.0	0.0	0.0	0.0	−9.0	−22.1	−40.7	−62.7	−90.8	−94.2	0.0
Cyl. No. 3	0.0	−5.2	−5.8	−4.7	−5.3	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Cyl. No. 4	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

E — CRANK MOMENTS FOR GIVEN ENGINE CYLINDERS

Cyl. No. 1	0	9640	12390	10590	7900	5700	3910	2770	2020	1410	840	390	0
Cyl. No. 2	0	0	0	0	0	0	−430	−1050	−1940	−2990	−4330	−4490	0
Cyl. No. 3	0	−250	−280	−220	−250	0	0	0	0	0	0	0	0
Cyl. No. 4	0	0	0	0	0	0	0	0	0	0	0	0	0

F — INERTIA TORQUES IN LB.-IN.

Cyl. No. 1	0	−1690	−2700	−2660	−1690	−250	1090	1920	2140	1890	1350	700	0
Cyl. No. 2	0	−700	−1350	−1890	−2140	−1920	−1090	250	1690	2660	2700	1690	0
Cyl. No. 3	0	−700	−1350	−1890	−2140	−1920	−1090	250	1690	2660	2700	1690	0
Cyl. No. 4	0	−1690	−2700	−2660	−1690	−250	1090	1920	2140	1890	1350	700	0

G — TOTALS OF CRANK MOMENTS AND INERTIA TORQUES (E + F)

Cyl. No. 1	0	7950	9690	7930	6210	5450	5000	4690	4160	3300	2190	1090	0
Cyl. No. 2	0	−700	−1350	−1890	−2140	−1920	−1520	−800	−250	−330	−1630	−2800	0
Cyl. No. 3	0	−950	−1630	−2110	−2390	−1920	−1090	250	1690	2650	2700	1690	0
Cyl. No. 4	0	−1690	−2700	−2660	−1690	−250	1090	1920	2140	1890	1350	700	0

H — TOTAL CRANK MOMENTS FOR ENGINE

0	4610	4010	1270	−10	1360	3480	6060	7740	7520	4610	680	0

Table VI.—Crank Moments Caused by Residual Connecting Rod and Gravity Forces.

CRANK ANGLES IN DEGREES

Cyl. No. 1	0	15	30	45	60	75	90	105	120	135	150	165	180
Cyl. No. 2	540	555	570	585	600	615	630	645	660	675	690	705	720
Cyl. No. 3	180	195	210	225	240	255	270	285	300	315	330	345	360
Cyl. No. 4	360	375	390	405	420	435	450	465	480	495	510	525	540

LB.-IN. TORQUE CORRECTIONS, ASSUMING WEIGHT OF ROD IS CONCENTRATED AT ENDS

Cyl. No. 1	0.0	−38.3	−67.7	−80.4	−71.5	−42.1	0.0	42.1	71.5	80.4	67.7	38.3	0.0
Cyl. No. 2	0.0	−38.3	−67.7	−80.4	−71.5	−42.1	0.0	42.1	71.5	80.4	67.7	38.3	0.0
Cyl. No. 3	0.0	−38.3	−67.7	−80.4	−71.5	−42.1	0.0	42.1	71.5	80.4	67.7	38.3	0.0
Cyl. No. 4	0.0	−38.3	−67.7	−80.4	−71.5	−42.1	0.0	42.1	71.5	80.4	67.7	38.3	0.0
Total	0.0	−153.2	−270.8	−321.6	−286.0	−168.4	0.0	168.4	286.0	321.6	270.8	153.2	0.0

LB.-IN. TORQUE CORRECTION FOR EFFECT OF WEIGHT OF RECIPROCATING PARTS

Cyl. No. 1	0.0	11.7	22.3	30.5	35.9	38.0	37.0	33.6	28.2	21.8	14.8	7.4	0.0
Cyl. No. 2	0.0	−7.4	−14.8	−21.8	−28.2	−33.6	−37.0	−38.0	−35.9	−30.5	−22.3	−11.7	0.0
Cyl. No. 3	0.0	−7.4	−14.8	−21.8	−28.2	−33.6	−37.0	−38.0	−35.9	−30.5	−22.3	−11.7	0.0
Cyl. No. 4	0.0	11.7	22.3	30.5	35.9	38.0	37.0	33.6	28.2	21.8	14.8	7.4	0.0
Total	0.0	8.6	15.0	17.4	15.4	8.8	0.0	−8.8	−15.4	−17.4	−15.0	−8.6	0.0

LB. IN. TORQUE CORRECTION FOR EFFECT OF UNBALANCED ROTATING PARTS

Cyl. No. 1	0.0	16.4	31.7	44.8	54.9	61.2	63.3	61.2	54.9	44.8	31.7	16.4	0.0
Cyl. No. 2	0.0	−16.4	−31.7	−44.8	−54.9	−61.2	−63.3	−61.2	−54.9	−44.8	−31.7	−16.4	0.0
Cyl. No. 3	0.0	−16.4	−31.7	−44.8	−54.9	−61.2	−63.3	−61.2	−54.9	−44.8	−31.7	−16.4	0.0
Cyl. No. 4	0.0	16.4	31.7	44.8	54.9	61.2	63.3	61.2	54.9	44.8	31.7	16.4	0.0
Total	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

NET EFFECT OF NEGLECTING ROD AND GRAVITY MOMENTS — (Algebraic sum of above total corrections)

0.0	−144.6	−255.8	−304.2	−270.6	−159.6	0.0	159.6	270.6	304.2	255.8	144.6	0.0

TOTAL CORRECTED TORQUE — (Algebraic sum of total crank moment as given in the last item Table V plus preceding item in this table)

0	4470	3750	970	−280	1200	3480	6220	8010	7820	4870	820	0

this force must be equal to the rate of work done on the crank by a moment M_g, that is,

$$P\frac{ds}{dt} = M_g\frac{d\theta}{dt}$$

$$\text{or } M_g = \frac{P}{w}\,v.$$

Substituting in Equation (3), (p. 324, August *P.E.*),

$$M_g = PrF_2 \tag{2}$$

the factors F_2 as given in Table II, (p. 324 August *P.E.*) are therefore applicable to both the expression for velocity and that for crank moment.

In determining the torque caused by the inertia force, item 2 above, the question arises of a proper division of the weight of the connecting rod between rotating and reciprocating weight. For this purpose it is generally assumed that the weight of the rod is concentrated at the piston pin and the crankpin. Letting l_1 be the distance from the center of the crankpin to the center of gravity of the rod, l_2 the distance from the center of gravity to the center of the piston pin, and W_r the weight of the complete rod, the concentrated rotating weight will be $W_r\ l_2/l$ and the reciprocating is $W_r l_1/l$. Letting W_p be the weight of the complete piston and pin and g the acceleration of gravity:

$$\frac{W_2}{g} = \frac{W_p + W_r l_1/l}{g}$$

This will be the mass the acceleration of which in the direction of stroke causes the inertia torque. By the same reasoning that was used in the development of Equation (2) above, this inertia torque is:

$$M_I = \frac{-\ W_2 arF_2}{g}$$

Substituting for a its value as in Equation (1) gives:

$$M_I = \frac{W_2 r^2 W^2 F_4}{g} \tag{3}$$

where $F_4 = F_2F_3$

(The equations for the correcting moment as given below are developed in Root, "Dynamics of Engine and Shaft" and Cormac, "A Treatise on Engine Balancing Using Exponentials".)

However, since the above presumes the impossible condition of a rod with massless midsection, some correction should be made on the inertia torque of Equation (3). The correcting value may be determined by adding some moment of intertia I_1 about its center of gravity, to the already assumed rod with weight concentrated at the ends, then making it dynamically equivalent to the actual rod whose moment of inertia about the center of gravity we may take as I. By this process:

$$I_1 = I - \frac{W_r l_1 l_2}{g}$$

The moment to correct for the force given above as item 3, is:

$$M_I = I_1 k^2 \omega^2 (u_2 \sin 2\theta + u_4 \sin 4\theta + \ldots) \tag{4}$$

$$u_2 = (\tfrac{1}{2} - \tfrac{1}{32}k^4 - \tfrac{1}{32}k^6 + \ldots)$$

$$u_4 = (\tfrac{1}{4}k^2 + \tfrac{1}{8}k^4 + \tfrac{1}{16}k^6 + \ldots)$$

In most cases it is found that the torque given by Equation (4) is small in comparison with the inertia torque caused by force (2) listed above, and may therefore be neglected.

The torque caused by force (4) as given above, may now be determined from the vertical component of the weight of the reciprocating parts. It is given by the equation:

$$Mw_2 = W_2 \cos \rho\ rF_2 \tag{5}$$

where ρ is the angle of the cylinder center line with the vertical.

The crank moment caused by the rotating weights, force 5, is given by the equation:

$$M_{w1} = W_1 \bar{r} \sin (\theta + \rho) \tag{6}$$

Here W_1 is the combined weight of the big end of the rod ($W_r\ l_2/l_1$), of the crankpin, and of the crank webs; and $\bar{r}$ is the distance from the center of gravity of W_1 to the center of the crankshaft. Like the correcting torque of Equation (4) the gravity moments of Equation (5) and (6) may usually be neglected, thus leaving it necessary to consider only the gas and inertia forces, items 1 and 2, to get a fairly accurate determination of the crank moment. However, in some vertical cross-head type engines where the weight of the reciprocating parts is great, the gravity torque may be considerable.

Tables I and II for calculating piston position, piston velocity and crank moment. (See p. 323 and 324, August *P.E.*) Table III and IV for calculating piston acceleration and for determining inertia torque. (See p.

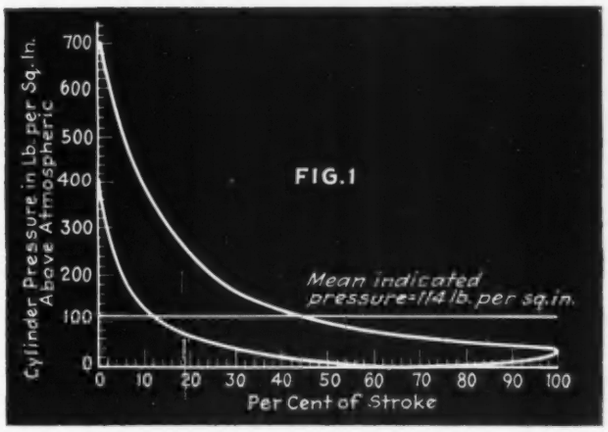
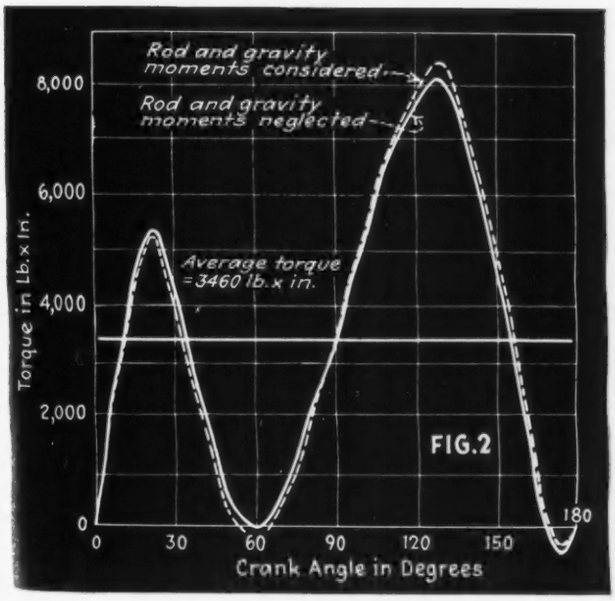


Fig. 1—Total effective pressures acting on the face of the piston can be readily calculated from the indicator diagram. Fig. 2—As shown by these curves, the errors resulting from neglecting the gravity and residual connecting rod moments are negligible.



363 and 364 in this number of *P.E.*)

(*Example*) The use of the tables is best demonstrated by an example. The following data are given for a vertical, 4-cylinder, 4-cycle diesel engine:

Bore	4½ in.	Weight of reciprocating parts (piston and upper end of the connecting rod)	12.35 lb.
Stroke	6 in.		
Speed	1,200 r.p.m.		
Firing order	1-2-4-3	Center to center length of connecting rod	13 in.

The indicator card shown in Fig. 1, is assumed for this engine. The calculations for the crank moment of this engine are carried out in Table V with the following explanations for each step:

(A) is the crank angle of each cylinder in degrees, measured from the top center position of the crank at the beginning of the expansion stroke. Starting with cylinder No. 1 at 0 deg. the calculations are carried out at 15 deg. intervals. Since this engine operates on the 4-stroke cycle the crank angles are measured up to 720 deg. and then repeated.

(B) is the position of the piston measured from top center, in per cent of the complete stroke. This is readily determined from Table I (page 323 August *P.E.*) by taking the crank radius r as 50 per cent of the stroke.

(C) is the cylinder pressure at the various crank angles, in pounds per square inch above atmospheric. Having found the piston positions (B) the cylinder pressure may be determined directly from the indicator diagram.

(D) = (C)×1×1×F_2, the crank moment of a single cylinder having a piston area of 1 sq.in. and a crank radius in in., the unit for the moment being lb.-in. See Equation (2). This torque is given for an engine of unit size in order that the gas torque of an engine of any bore and stroke, operating on the same indicator card and having nearly the same ratio of rod to crank, may readily be determined.

(E) is the crank moment due to the gas force on the piston of a single cylinder of the engine in question. It is equal to $\frac{\pi}{4}(4.5)^2 \times 3 \times (\text{Item D}) = 47.7 \times (\text{Item D})$

(F) is the inertia torque of a single cylinder in pounds times inches, according to Equation 3. It is equal to:

$$-\frac{12.35}{386} \times (3)^2 \times \left(\frac{2\pi \times 1200}{60}\right)^2 \times F_4 = -4550F_4$$

(G) equals (E) + (F). It is the net gas and inertia torque of a single cylinder of the engine in lb.-in.

(H) is the crank moment developed by the combined 4 cylinders and expressed in lb.-in. For example, at the instant cylinder No. 1 is at 60 deg., No. 2 is at 600 deg., No. 3 is at 240 deg., and No. 4 is at 420 deg. crank angle. Therefore, the combined moment is 6210-2140-2390-1690 = —10 lb.-in.

In order to show the influence of the residual force (3), as given in Equation (4), on the crank moment of the engine, a few further calculations must be carried out. For these calculations the following additional data are given:

Weight of the unbalanced crank webs and crankpin	23.22 lb.
Distance from the center of gravity of the unbalanced crank webs and crankpin to the center line of the crankshaft	1.798 in.
Weight of the complete connecting rod	10.8 in.
Weight of the lower or rotating end of the connecting rod	7.2 lb.
Moment of inertia of the complete connecting rod about its center of gravity	0.860 lb.— in. x sec.²

From the foregoing data $l_1 = \frac{13 \times 3.6}{10.8} = 4.33$ in.

and $l_2 = \frac{13 \times 7.2}{10.8} = 8.67$ in.

Hence $l_1 = 0.860 - \frac{10.8 \times 4.33 \times 8.67}{386} = -0.191$ lb.-in.sec.²

From Equation (7)

$$u_2 = \left[\frac{1}{2} - \frac{1}{32} \times \left(\frac{1}{4.33}\right)^4 - \frac{1}{32} \times \left(\frac{1}{4.33}\right)^6 + \ldots\right] = 0.4999$$

$$u_4 = -\left[\frac{1}{4} \times \left(\frac{1}{4.33}\right)^2 + \frac{1}{8}\left(\frac{1}{4.33}\right)^4 + \frac{1}{16}\left(\frac{1}{4.33}\right)^6 + \ldots\right] = -0.0137$$

$$u_6 = \left[\frac{3}{32} \times \left(\frac{1}{4.33}\right)^4 + \frac{3}{32} \times \left(\frac{1}{4.33}\right)^6 + \ldots\ldots\ldots\right] = 0.0003$$

and

$$M_r = 0.191 \times \left(\frac{1}{4.33}\right)^2 \times \left(\frac{2\pi \times 1200}{60}\right)^2 (0.4999 sin2\theta - 0.0137 sin4\theta + 0.0003 sin6\theta)$$

$$= -80.4 sin2\theta + 2.2 sin4\theta \qquad (7)$$

By Equations (5) and (6) the gravity torques of a single cylinder of the engine are:

$$M_{W2} = 12.35 \times 1.000 \times 3 \times F_2 = 37.05 F_2 \qquad (8)$$

$$\text{and } M_{W1} = 30.42 \times 2.08 \, sin\theta = 63.3 \, sin\theta \qquad (9)$$

Note: $W_1 = 23.22 + 7.2 = 30.42$ lb.

$$\text{and } r = \frac{23.22 \times 1.798 + 7.2 \times 3}{30.42} = 2.08 \text{ in.}$$

Equations (7), (8) and (9) have also been carried out for a 4-cylinder engine in a tabular form, the results alone being shown in Table VI. These were added to the values of H given in Table V and plotted in Fig. 2, together with the values of H alone in order to show the error in neglecting the residual connecting rod and gravity moments. It is evident from these curves that there is usually little error in the neglect of the rod and gravity torques of the usual reciprocating engine with trunk type pistons.

From the foregoing tables and calculations we may conclude:

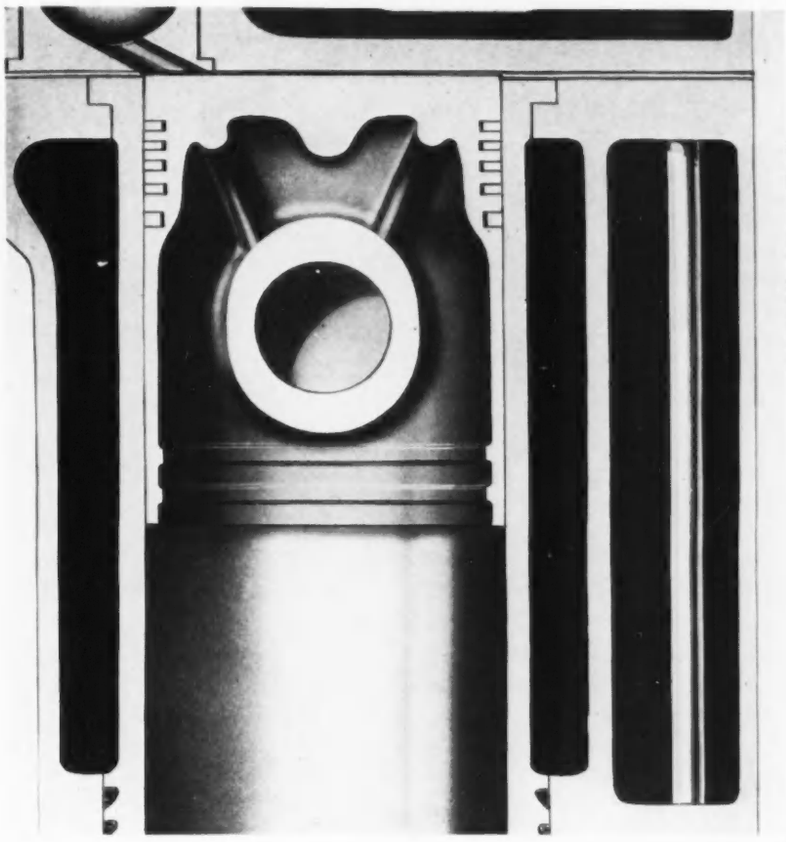
1. The positions of a piston connected to a crank of unit radius have been determined for a number of ratios of length of connecting rod to crank and for a number of crank angles. From these values the position of the piston of a reciprocating machine having any crank length may be determined. The tabulated values are also useful in the analysis of indicator cards.

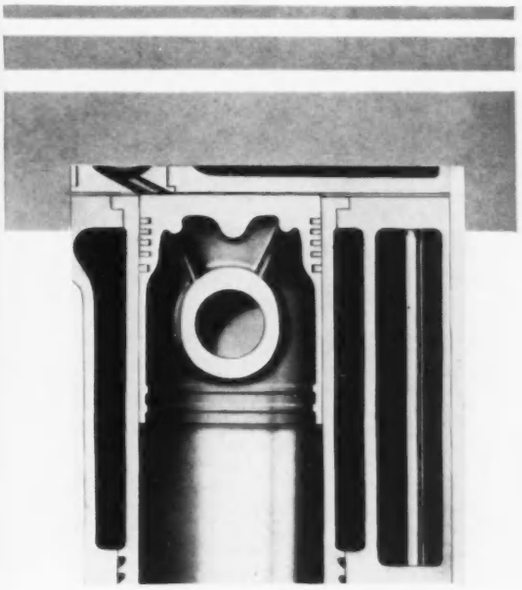
2. The velocities and accelerations of a piston have likewise been tabulated for an engine of unit crank radius and rotating at unit angular velocity. From these tables the instantaneous piston speed and acceleration may be calculated for a machine of any size and speed.

3. The factors tabulated for the piston velocity may also be used for the determination of crank moment by simply multiplying by the crank radius and the gas or inertia force on the piston.

4. By multiplying the velocity (or crank moment) factors by the acceleration factors a table has been set up whose values, when multiplied by the mass of the reciprocating parts, the crank radius squared, and the angular velocity squared, gives the inertia torque.

5. For such purposes as the determination of required flywheel sizes, consideration of only the gas pressures and inertia forces on the piston usually are necessary in calculating crank moments. However, in some low-speed machines of the vertical, cross-head type the gravity forces on the reciprocating parts may be so great that they should be added to the gas force on the piston.





Proportioning Sizes of Similar Parts

Doubling physical dimensions does not necessarily increase all properties in the same ratio, some characteristics remain constant and others vary in a definitely predictable manner

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MACHINE designers are frequently confronted with the proposition of determining the relationship existing between the characteristics of machines different in size yet very similar in geometrical proportions and operating functions.

Changing conditions sometimes make it necessary to replace a machine which has been satisfactory and efficient with a newer or larger model to meet a new demand. It then becomes the designer's job to lay out a machine of new capacity having all the merits of the old one. Competition, styling, improvements in current practice, or the use of new materials may make it desirable to incorporate additional features of value in the new machine, any of which would complicate the task.

Mathematical analysis is not always a convenient method of investigating the properties of very large complex structures, nor is it to be solely relied upon where the consequences of possible failure would be disastrous. Models, rarely complete in all details, are often made to scale and then tested exhaustively to secure data that can be applied to the design of the full size structure. Using a different material for the model will sometimes yield useful knowledge as to how to proceed and what to avoid. Photo-elastic investigations have been recognized in many fields of design as a very valuable method of studying effects of contour on stress distribution. In applying any information gained from tests or past experience the designer must check from two angles, he must determine those vital characteristics which remain constant with change of proportion or scale, and those which vary in a definite predictable manner.

The study of the related characteristics possessed by objects of similar geometric proportions is very old, it is known to science as the Principle of Homogeneity or the Principle of Similitude. Galileo (1564-1642) when investigating accidents in the Venetian shipyards, observed the fallacy of assuming that doubling the size doubled all other properties and characteristics. He found that designers were specifying large beams with proportions geometrically similar to successful small ones. As is usual with such mal-practice the structure was satisfactory if the original factor of safety was large

or the increase in size was small. However, when attempts were made to build still larger the end was disastrous.

Present day designers generally do not assume that geometrically similar bodies have similar characteristics, nor do they blindly "scale up" or "scale down." If one is redesigning a familiar machine that can be inexpensively made and tested to destruction in a short time, it might be permissible to determine the new proportions and configurations of the new machine or part by exercising his judgment or experience. This procedure can only be followed in a very limited number of cases, and is successful only when based on long years of practical experience. An additional check on important dimensions is desirable in every case as an assurance against a failure which would involve loss in time or property.

To avoid ambiguity and consequent misunderstanding of terms which will be used in this discussion when comparing similar machines, some definitions are included.

SIZE: The size of an object is its measurements expressed in dimensions; doubling the size doubles all three dimensions—length, breadth and thickness.

CONFIGURATION: The configuration of an object is the physical arrangement of its structure, embracing its shape, contour and the harmony existing between its component parts.

CHARACTERISTICS: These can be conveniently grouped into:

1. The physical and chemical properties; such as weight, surface area, strength, elastic elongation, hardness, resilience, resistance to corrosion, heat and electrical conductivity, cost of production, and many others.

2. The functions of the object; meaning the uses to which it is put, the part it plays in relation to other elements in transmitting motion, loads or forces, or what it can do under certain imposed conditions.

PROCEDURE: For the purpose of maintaining orderly arrangement and method when investigating the problems presented by similitude in machine design, the following sequence is suggested.

1. Determination or assumption of the increased or decreased capacity desired for the new machine.

2. Listing the important characteristics that will be common to both machines.

3. Listing the characteristics that will probably differ with a change in size. All doubtful items should be included under this heading.

4. Determination of the increased size of the new machine if the configuration is unchanged.

5. Determination of the exact relationship of the variable characteristics if the configuration is unchanged.

6. Listing the characteristics that have changed:—
   (a) Advantageously,
   (b) Disadvantageously,
   (c) Without effect.

7. Discussion of step (6).

8. Modification of configuration, material, cost, operating conditions, life of machine and like factors for the purpose of getting a working compromise.

EXAMPLE: A gasoline engine is to be redesigned with the objective of increasing the capacity 800 per cent. This is a very great difference and is chosen simply for ease in calculating and also not to detract attention from the method. Data in regards to the old successful piston is assembled from which as a basis the new one is to be designed.

STEP 1. The increased capacity is to be 800 per cent.

STEP 2. The characteristics common to both the new and old pistons and engines are:

(a) The r.p.m. of crankshaft.

(b) Shape of indicator diagram, as well as the values of the mean effective unit pressure and the maximum unit pressure.

(c) Ratio of the length of the connecting rod to that of the stroke.

(d) The specific gravity of material.

(e) Thermal efficiency of the engines.

STEP 3. The characteristics likely to change are:

(a) Volume of piston,
(b) Weight of piston,
(c) Weight per horse-power,
(d) Inertia force on connecting rod bolts,
(e) Unit stress in connecting rod bolts,
(f) Area of surface in contact with cylinder,
(g) Side thrust of piston,
(h) Unit stress between piston and cylinder,
(i) Piston speed,
(j) Friction between cylinder and piston,
(k) Mechanical efficiency,
(l) Operating temperature of piston crown.

STEP 4: Determination of linear dimensions:

(Note: Capital letters designate and refer to large machine, and small letters will be used to indicate small machine.)

Capacity is to be increased 8 times or

$$H.P. = 8\ h.p., \quad (1)$$

$$PLAN = 8plan, \quad (2)$$

$$33{,}000 = 33{,}000$$

where P = mean effective pressure,
L = length of stroke,
A = area of piston,
N = revolutions per minute.

Cancelling out the constants Equation (2) becomes

$$LA = 8\ la \text{ or } LD^2 = 8\ ld^2 \quad (3)$$

using a scale ratio factor k_s to show the relations $L = k_s l$, and $D = k_s d$, and substituting these expressions in Equation (3)

$$(k_s l)\ (k_s d)^2 = 8ld^2$$

$$k_s^3 = 8 \text{ or } k_s = 2$$

This means the conditions expressed in Equation (2) can be attained by doubling the linear dimensions, in this case the piston dia. and the length of stroke.

STEP 5. Relationship between variable characteristics.

(a) Volume of piston displacement

$$V = k_v v,$$

where k_v = volume ratio factor,

$$LD^2 = k_v ld^2,$$

$$\text{or } (2l)(2d)^2 = k_v ld^2,$$

whence $k_v = 8$, or the volume will be increased 8 times.

(b) Since the specific gravity is constant the weight will be increased 8 times.

(c) Weight per horse-power will be unchanged since the weight and the horse-power have both been increased 8 times.

(d) Inertia force on connecting rod bolts.

$F = 0.00034\ WRN^2[cos.\theta + (R/C)cos2\theta]$

$f = 0.00034\ wrn^2[cos\theta + (r/c)cos2\theta]$

$F = k_f f$, (4); where F = inertia force,

θ = angle crank has turned from top center,

R = crank radius, C = length of connecting rod,

W = weight of reciprocating parts,

k_f = inertia ratio factor, then

by cancelling of constants and substituting, Equation (4) becomes $WR = k_f wr$ or $(8w)(2r) = k_f wr$, whence $k_f = 16$ or the inertia load on connecting rod bolts is increased 16 times.

(e) Unit stress in bolts.

Increase in bolt area is 4 since the diameter is increased by 2 being a linear dimension. Unit stress ratio factor

$$k_u = \frac{\text{increase in inertia load}}{\text{increase in bolt area}} = \frac{16}{4} = 4$$

or unit stress in connecting rod bolts is increased 4 times.

(f) Increase of surface area in contact with cylinder;

$$S = k_c s \qquad (5)$$

where S = surface area of piston

H = length of piston

k_c = surface ratio factor

Equation (5) can be developed to this form

$$\frac{\pi DH}{2} = k_c\left(\frac{\pi dh}{2}\right),$$

whence $k_c = 4$, or surface area is increased 4 times.

(g) Increase in side thrust.

$$0.173D^2P^1(R/C) = 0.173k_t d^2 p^1(r/c) \qquad (6)$$

where P^1 = total pressure on piston

k_t = thrust ratio factor

Since $P' = PA = (\pi/4)\ D^2P$; also $p' = Pa = (\pi/4)\ d^2p$, and $P = p$; it follows that $P^1 = 4\ p^1$. Equation (6) becomes through cancellation and substitution $(4d^2)\ (4\ p^1) = k_t d^2 p^1$, and $k_t = 16$, or the side thrust is increased 16 times.

(h) Unit stress between cylinder and piston.

Ratio increase factor

$$k_b = \frac{\text{increase in side thrust}}{\text{increase in areas}} = \frac{16}{4} = 4,$$

or unit stress between cylinder and piston is increased 4 times.

(i) Piston speed; $V = k_p v$, (7)

where V = velocity of piston

k_p = velocity ratio factor

Equation (7) becomes $2\ LN = 2k_p ln$, and $2l = k_p l$, and $k_p = 2$, or the piston speed is doubled.

(j) Friction between cylinder and piston;

$$W^1 = k_w s^1, \qquad (8)$$

where W^1 = work of friction,

k_w = friction ratio factor.

Let F^1 = force friction

V = velocity

μ = coefficient of friction,

S^1 = side thrust of piston

then $F^1 V = k_w f^1 v$, or $S^1 \mu V = k_w s^1 \mu v$,

but $S^1 = 16s^1$, and $V = 2v$, therefore

$(16\ s^1)\ (2v) = k_w s^1 v$, and $k_w = 32$

work lost in friction increases 32 times.

(k) Mechanical efficiency;

$$E = k_m e, \qquad (9)$$

where E = mechanical efficiency

k_m = efficiency ratio factor.

Let B = output, and G = loss, then

$\frac{B}{B+G} = k_m\left(\frac{b}{b+g}\right)$, or $\frac{8b}{8b+32g} = k_m\left(\frac{b}{b+g}\right)$, and $k_m = \frac{b+g}{b+4g}$, assume the small engine lost 5 per cent of input, then

$k_m = \frac{100+5}{100+4\,X5} = 87.5$ per cent, or the large engine would be 87.5 per cent as mechanically efficient as the small engine.

(l) Operating temperature;

$$T = k_r t, \qquad (10)$$

where T = difference in temperature between center and edge of piston, and k_r = temperature difference ratio factor.

Let H^1 = quantity of heat released,

C = thermal conductivity,

Q = piston thickness.

Equation (10) becomes

$$\frac{H^1(D/2)^2}{4CQ} = k_r\left(\frac{h^1(d/2)^2}{4cq}\right),$$

$$\frac{8h^1(2d/2)^2}{2q} = k\left(\frac{h^1(d/2)^2}{q}\right), \qquad (11)$$

And $k_r = 16$, or the temperature difference is increased 16 times.

STEP 6. One characteristic has changed advantageously: capacity increased 8 times.

The following characteristics have changed disadvantageously.

(a) Unit tensile stress in connecting rod bolts increased 4 times.

(b) Unit bearing stress between cylinder and piston increased 4 times.

(c) Piston speed increased 2 times.

(d) Mechanical efficiency decreased.

(e) Operating temperature difference of piston increased 16 times.

STEP 7. Examination reveals that two important unit stresses have increased. Perhaps the area of the connecting rod bolts can be increased or a stronger material used for them. Another scheme would be to reduce the inertia load by using a lighter material for the piston. The increased unit pressure between piston and cylinder presents a lubrication problem.

The increase in the operating temperature of the piston is disturbing, since to keep this down to a value that will eliminate the possibility of pre-ignition, both the configuration and material will have to be changed.

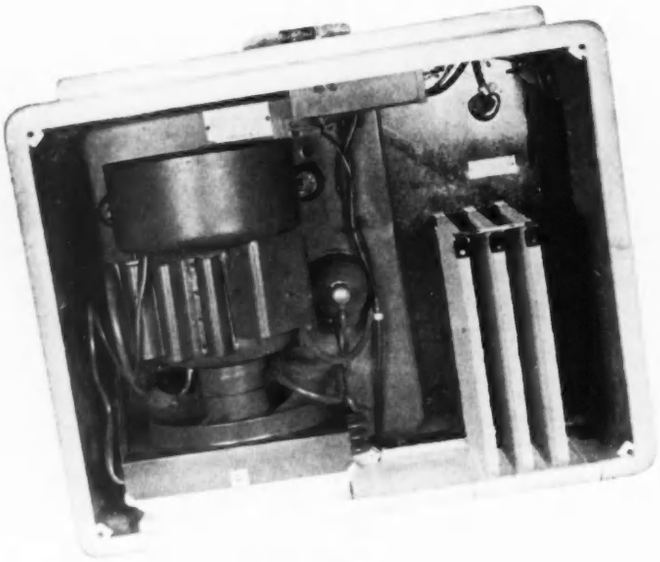
STEP 8. Temperature increase if thickness of piston head is increased 8 times and aluminum is used instead of iron, then because of the difference in thermal conductivity, the coefficient for aluminum being 3 times that for iron, Equation (11) becomes

$$\frac{8h^1(2d/2)^2}{4(3C)\ (8q)} = k_r\left(\frac{h^1(d/2)^2}{Cq}\right).$$

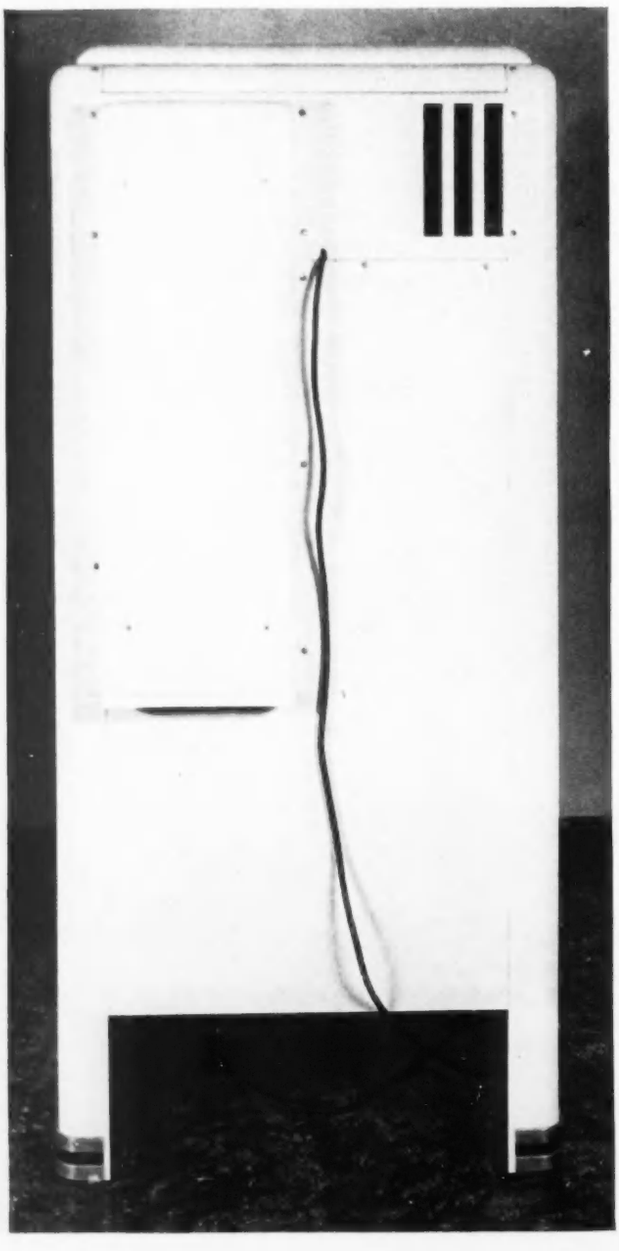
or $k_r = 1\frac{1}{3}$ which may be acceptable. Further modifications will have to be made to get a working compromise, and the designer must exercise judgment as what increases to permit in unit stress, temperature and other vital factors.

The method outlined depends entirely on the designer's resourcefulness in expressing the relationship between the factors involved, and is not a substitute for design skill or engineering ability. If the wrong formula is used, or if the correct one is wrongly interpreted, the results will be in error. However, in making use of the outline the designer is brought to a realization of the importance of size, configuration and characteristics of similar machines.

Baffles



Successive design of ducts and baffles with tests for ventilation and relative damping effects by which a silencing system in an electric refrigerator was achieved



THE suppression of noise as a refinement in domestic products receives as much attention as mechanical design and appearance appeal. With each successive model attempts are made to make operating noise less noticeable. In the household appliance field probably the refrigerator serves over the greatest period of time. Its quiet performance is largely the result of designing a mechanism that generates as little vibrational energy as possible. Since sound waves have power or energy, the logical attack on noise elimination is to de-energize the waves by some means or other. One method is the use of a sound insulating substance of fibrous composition having the fibers stacked loosely. When the pressure wave is projected against such material, its energy is used up by the working of the fibers and is dissipated in the form of heat.

The development of a noise baffling system, test apparatus and test methods used by the Westinghouse company are described here. Since noise is relative, it was necessary to establish what the average ear considered a quiet machine. This was done by placing a number of refrigerators in a room insulated from exterior noises, and having several people listen to them and decide which one was most acceptable. This machine was then used as a standard from which to work, and was set up in a soundproof room. A noise meter was used to record on a dial the amount of noise produced by the machine at various points around the cabinet.

The refrigeration cabinet is divided into the food storage compartment, from which heat is picked up by the refrigerant circulating in a coil, and the box-shaped refrigerator-unit compartment which houses the motor and machinery and where the refrigerant gives up or dissipates heat. This heat is removed from the compartment by fan circulated air. A problem, therefore, was to provide adequate air circulation with sufficient sound baffling so that the openings in the compartment for the circulation of air would not permit the egress of noise. Several limitations were imposed. The cabinet could not stand out from the wall more than $2\frac{1}{2}$ in. and care had to be taken to prevent the recirculation of the ingoing and exhaust air. Appearance could not be jeopardized by cutting louvers in the front or sides and the particular cabinet design prevented the exhaust air from going out the top of the compartment.

As the refrigeration unit took up most of the space from the front to back of the compartment, as shown in

Solve a Noise and Ventilating Problem

S. C. EAKER

Development Engineer, Westinghouse Electric & Manufacturing Company

Fig. 1, it was apparent that an extension or duct would have to be built to take the air in and exhaust it.

The earlier models took cool air in from the floor, circulated it over the unit and then turned it out and up as indicated in Fig. 1, which was an advantage for cooling purposes. The openings in the compartment provided adequate ventilation but the noise was greater than desired.

Tests indicated that by lengthening the intake duct and lining it with sound absorbing material, noise from this opening could be reduced. To lengthen the exhaust duct upward to take care of noise would be impossible from an appearance standpoint. Therefore, the outgoing air had to be turned down and this passageway also lined with sound absorbing material which was also a heat insulator. A spreader was incorporated in the openings at the end of the duct to prevent recirculation

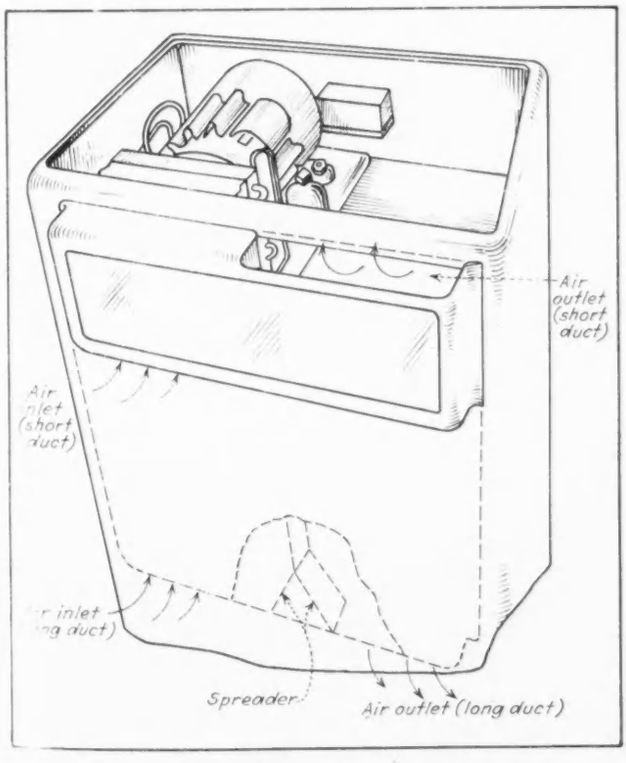


Fig. 1—First scheme using a short duct is shown in full lines with air entering underside and flowing out of top side. Dotted lines indicate improved duct extended with spreader

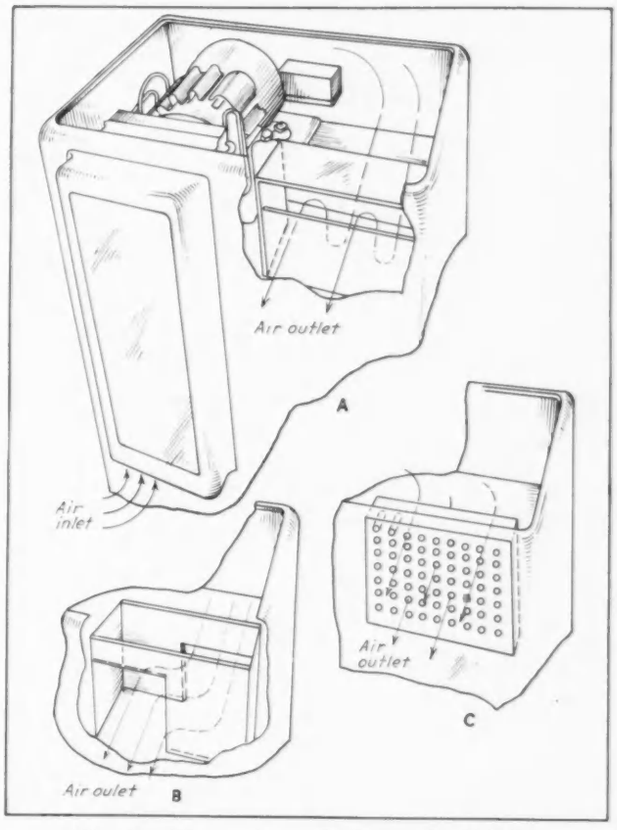


Fig. 2—Progressive design of baffling arrangement in attempts to break up sound waves (A) Simple two-turn baffle. (B) Elementary box with offset inlet and outlet (C) Partition with large number of small holes

of the exhaust air. It was also found that a metal connection from this duct to the condenser permitted too much of the higher frequency noise to be transmitted to, and amplified by, the cabinet panels. The substitution of a cloth connector eliminated this objectionable feature. In addition the walls of the motor compartment were lined with sound insulation which brought the noise down to an acceptable level.

However, development tests revealed that the ventilating scheme just described and used on the 1934 model refrigerator could be improved

Air velocity through the system previously described was 440 ft. per min. This would be adequate for cooling purposes if there were no chance for recirculation

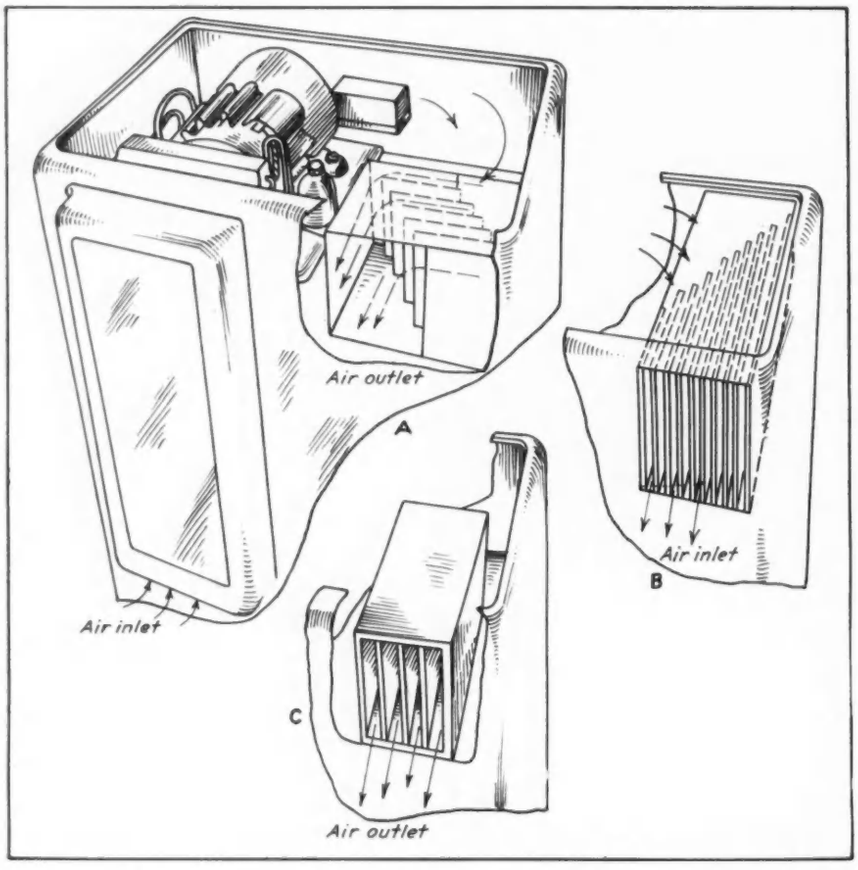


Fig. 3—Development of parallel passage baffles (A) How pieces of heavy cloth were used. (B) Multiple plates in graded lengths. (C) Next step was larger openings and fewer plates

of the exhaust air. When the cabinet was used in an alcove installation, particularly one so small that there was no chance for air to come from around the sides of the box, recirculation was encountered. A long duct for the inlet, lined with sound absorbing material was fundamentally good, as it would take air from the floor where it was comparatively cool. The noise emanating from this opening in the unit compartment was projected against the sound insulation of the duct, some of it was absorbed and some reflected against the metal back of the cabinet which deflected it again to the insulating lining. By making the duct long enough, this construction was effective in coping with the noise.

The exhaust arrangements presented another problem of devising a scheme with little restriction as far as air was concerned but considerable noise reducing capacity. When the outlet side was left open without any obstruction the air velocity was 610 ft. per min. This, of course, was the figure to strive for in building a baffling system. Working on the theory that a considerable surface material would absorb the sound, labyrinths were devised through which the outgoing air passed. Fig. 2*A* shows one scheme. The degree of noise was acceptable but the air velocity was only 505 ft. per min. Fig. 2*B* shows another scheme with which the air velocity was increased up to 526 ft. per min. but the noise was also greater. It was thought by putting the air through a large number of small holes, as in Fig. 2*C*, the sound waves would be broken up and the air streams would not be affected greatly. The sample tried was satisfactory as a noise baffle but the air delivery was cut to 405 ft. per min. showing that the holes would have to be considerably larger. In view of this, a scheme of parallel passages, made from insulating material, was evolved. Fig. 3*A* shows a form made of parallel pieces of heavy cloth through which the air passed. The noise was satisfactory and the delivery rate, 565 ft. per min., was good.

The scheme shown in Fig. 3*B* was tested next. The noise was less with this than with anything previously tried and the air velocity was up to 600 ft. per min. From this to the scheme shown in Fig. 3*C* and then to the final design, Fig. 4, was a brief step. The degree of noise was acceptable. The air velocity was up to 610 ft. per min. which was the predetermined objective. The length of the parallel plates needed was determined by test. A balance had to be established between the noise and the air delivery. The longer the pieces were the less the noise, but the air velocity was also less as the restriction was increased. This design for the ventilation of the motor compartment and scheme for noise baffling produced a very efficient refrigerating unit. It provided the maximum amount of cooling air, eliminated any chance of recirculation and lowered the noise to an acceptable level.

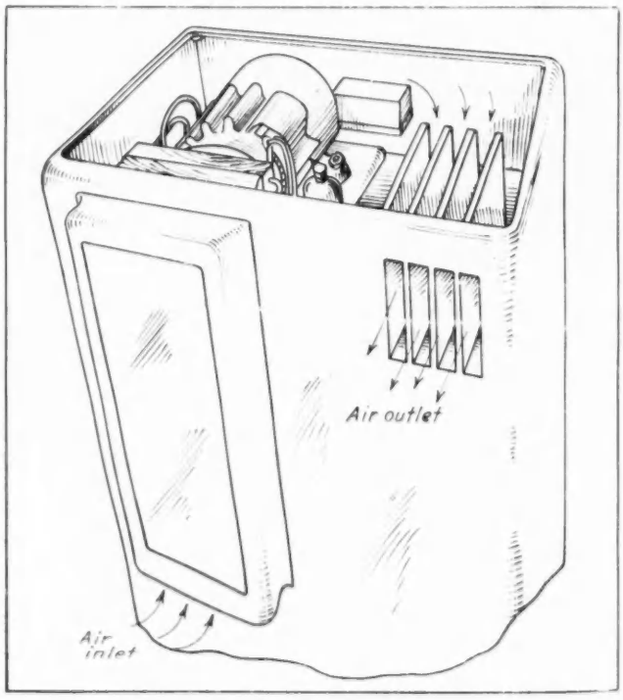


Fig. 4—Final design was balanced to compromise noise and air delivery

QUESTION AND COMMENT

Dial Indicator for Adding and Subtracting Common and Mixed Fractions

H. O. NEKOLA
Cleveland, Ohio

●In drafting and checking where it is continually necessary to add or subtract fractional dimensions, the device shown in the illustration was made for the purpose of saving time and estimating errors. It consists of a base upon which is marked a series of divisions each division representing 1/64 in. and the decimal equivalent of each fraction. The base is recessed in which a rotating dial is pivotally mounted, the upper face being flush with the top surface of the base, and is divided into small sectors, each sector corresponding in length to one division of the outer scale. In each sector is drilled a small hole into which the point of a pencil can be placed for rotating the dial and one has an index mark.

To add up a series of fractions such as $\frac{1}{8}$, 3/64, 21/32 and $\frac{5}{16}$, the instrument is set as shown with the index mark opposite the 1.00 division on the base. The pencil point is placed in the hole on this sector and by this means the rotating disk is moved in a clockwise direction until the index comes opposite the $\frac{1}{8}$ figure. The pencil point is then placed in the hole which has then come opposite the 1.00 division and the dial again moved in a clockwise direction until the pencil point comes opposite the 3/64 division. The same is done for 21/32 and $\frac{5}{16}$ figures and when completed, the marker will be found to have made one complete revolution and to have stopped opposite the 9/64 division of the stationary scale indicating that the sum is 1 9/64 in.

Because the fractions often add up to more than one inch, a counting arrangement is incorporated which consists of a small toothed dial operated by a single tooth on the large dial. At each revolution of the large dial the single tooth engages one of the teeth on the small toothed dial to move that member part of a revolution. This dial has 11 teeth and 11 divisions and is mounted on a pivot and located within the recess in the base. A hole is cut through the scale on the base in such a location that one of the numbers on the toothed dial always shows through this hole. The number appearing through the hole represents the number of revolutions the small dial has made, or the number of whole inches in the addition, while the index marker indicates the fractional part of the inch.

If a dimension is to be subtracted from a sum, the pencil point is merely placed in the hole opposite the dimension which is to be subtracted and the large dial moved in a counter-clockwise direction until the pencil comes opposite the 1.00 section on the stationary dial, the index marker indicating the result of this subtraction. For instance, if 39/64 is subtracted from the 1 9/64 sum the pencil point would be placed in the hole opposite the 39/64 division. On moving the large dial counter clockwise direction until the pencil is opposite the 1.00 division the index marker arrives at the 17/32 division. During this last operation the single tooth on the large dial will have moved backwards and will have carried the small dial backwards until the O mark shows through the hole 2. In this instance it will be the O mark in the inner of the two circles of numbers on the tooth dial which will show. This comes about because in subtracting the single tooth contacts the opposite side of the small dial tooth.

Hydraulic Pressure Drop

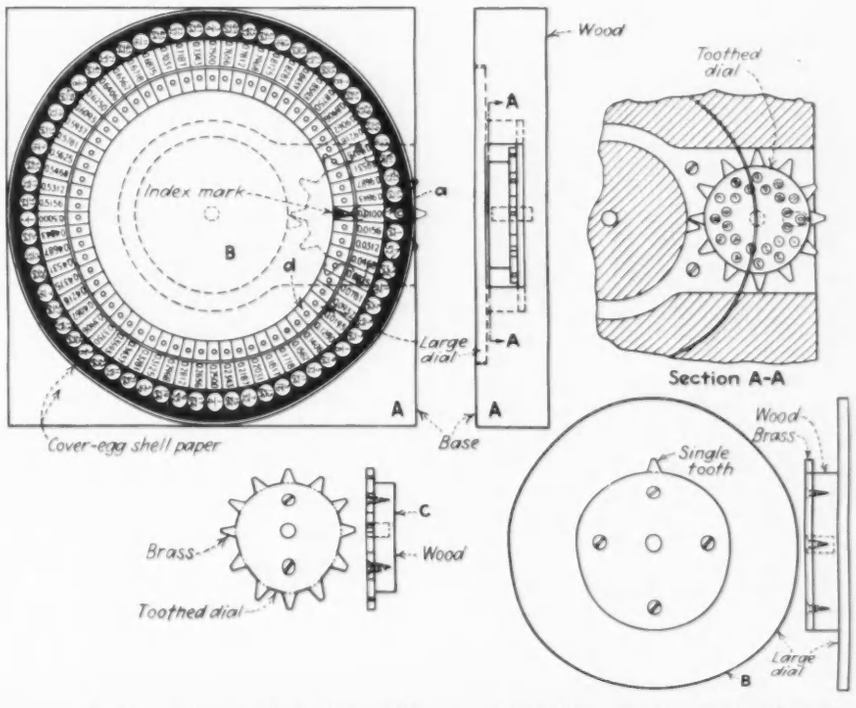
To the Editor:

●I would like to call your attention to an error in the four equations in the first column on page 125 of your April, 1935, number. None of the constants seem correct. And what is the basis for the first equation?

—E.N.B.
Lancaster, Ohio

To the Editor:

●E.N.B. is partially correct in his suggestion, but this only applies to the second equation which should read $P = 128\,\mu LQ/\pi d^4$. The first equation



Easily made calculator for adding or subtracting common or mixed fractions found useful for dimensioning or in checking drawings

is the well known Poiseuille equation which is found in most standard text books on hydraulics. As is stated in the paragraph preceding this equation, it applies only where the flow is truly streamline or sinusoidal. For turbulent flow a different relationship exists.

The other equations are derived directly from the Poiseuille equation by simple substitution and transformation. The third and fourth equations are correct inasmuch as the constant has been calculated taking π into consideration and is obtained as follows: $128/(144 \times 60 \times 3.1416)$. Therefore, the sign π should be inserted only into the denominator of the second equation.

—HANS ERNST
Research Engineer
Cincinnati Milling Machine Company

Vertical Shaft Bearing Lubrication

To the Editor:

●In the March issue, Fred Adams asks for a suggested method for automatically lubricating vertically mounted ball bearings. The device shown in the accompanying sketch may solve his problem. An oil reservoir, *A*, is provided below the lower bearing and a tube *B* is placed in a keyslot in the shaft; the lower end of the tube being bent out and down into the oil reservoir and turned forward in the direction of motion, so that it acts as a scoop when the shaft rotates. When a certain speed is reached, depending on the constants of design, oil is forced up through the tube and discharged over the upper bearing at *C*; flowing down through both bearings and back to the reservoir.

This device will pump only above a certain speed. Oil will circulate in the system when the velocity pressure, because of the motion of the scoop in the oil, exceeds the sum of the static head of the oil in the tube, and the centrifugal back pressure caused by the difference between the radii r_1 and r_2.

m = specific mass of the oil
n = shaft rev. per sec.
v = velocity of an elemental particle of oil in tube at radius r in ft. per sec.
g = gravitational constant
h = height of oil in the tube, in ft.
v_1 = velocity of tip of scoop in ft. per sec.

If the oil were stationary in the reservoir, the pumping head due to the action of the scoop will be

$$\tfrac{1}{2} m v_1^2 = \tfrac{1}{2} m (2\pi n r_2)^2$$

The oil will not, however, be stationary, but will have a certain forward motion which will reduce the effective pumping velocity by a factor R whose value depends on the design of the oil reservoir and usually lies between 0.5 and 0.9. As will be shown, the greater the value of R, the more favorable are the operating conditions. A high value of R is obtained by designing the oil reservoir in such a way as to retard as much as possible the circulation of oil in it. This may be accomplished by making the reservoir deep and narrow and providing ribs on the walls. Also, the submerged rim of the tube should be sharpened to a knife edge. The actual pumping pressure becomes:

$$\tfrac{1}{2} m (2\pi n r_2 R)^2$$

This pressure must overcome the static head, mgh, of the oil in the tube, plus the centrifugal back pressure, which is expressed by the integral,

$$m \int_{r_1}^{r_2} \left(v^2 dr/r \right)$$

The complete condition for circulation of the oil then is

$$\tfrac{1}{2} m (2\pi n r_2 R)^2 > mgh + m \int_{r_1}^{r_2} [(2\pi n r)^2 dr/r]$$

By integrating and simplifying the condition for circulation is obtained as:

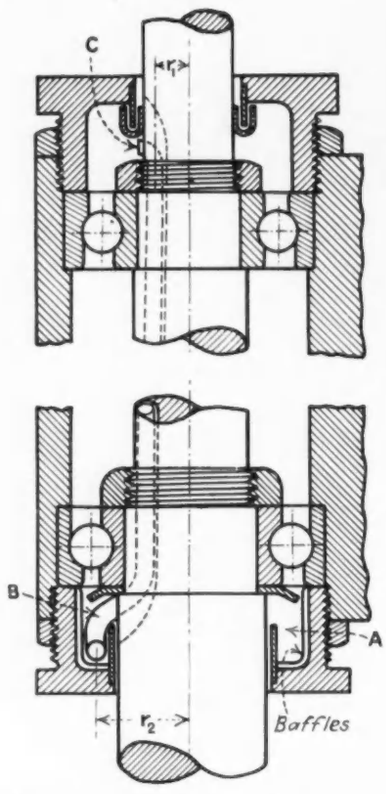
$$n > \sqrt{gh} \Big/ \sqrt{2\pi^2 [r_1^2 + (R^2 - 1) r_2^2]}$$

If the denominator of the fraction to the right is negative, the value of n becomes imaginary. For a real value of n (that is, for circulation to take place) it is necessary that

$$r_1^2 + R^2 r_2^2 > r_2^2$$

or

$$r_1 \Big/ r_2 > \sqrt{1 - R^2}$$



Lubricant is picked up by rotating tube in slotted shaft and forced upwards discharging over upper bearing.

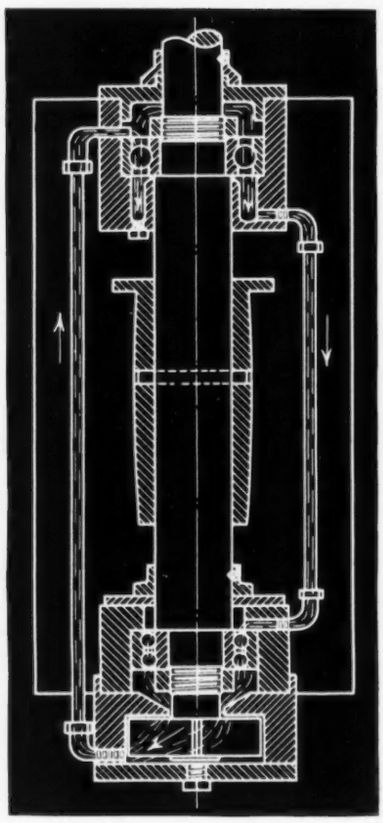
Other methods for doing the same work are available. Though less critical in design, most of them are, however, more complicated and costly in construction.

—LEO G. HALL
Allegan, Mich.

Bearing Lubrication for Vertical Shafts

To the Editor:

● With reference to the question of automatically oiling two bearings, 6 in. apart on a vertical shaft as requested by Fred Adams, a design of a high-speed woodworking shaper spindle is shown here. A constant-level oil well



Oil is pumped through tube to upper bearing chamber from where it flows by gravity through both upper and lower bearings.

is provided below the lower bearing, and a small impeller attached to the lower end of the spindle and runs in the oil. When the spindle is rotating, the oil is forced through the tube on the left to the top of the upper bearing from which point it drains through the bearing cage and into the annular channel below the bearing. From this channel the oil is drained by the tube, on the right, to the upper side of the lower bearing, through the bearing raceways and back to the oil chamber.

In this design there is a constant

gravitational flow of oil through both bearings, the lower bearing not relying on splash to lubricate it from below. A constant level device is incorporated so that the depth to which the pump impeller is submerged is closely governed to prevent churning and consequent heating of the oil.

—JOHN E. HYLER
Peoria, Ill.

Piston Rings versus Cup Leathers

To the Editor:

●I would like to obtain information relative to the use of piston rings and cup leathers in hydraulic cylinders. What are the limiting factors of each type of sealing agent regarding reciprocating speed, frequency of reversal and holding power? How small and how large can each be made to give efficient operation? What kind and grade of material is used in both rings and leather?

—JOHN BEST
Baltimore, Md.

To the Editor:

●Referring to the subject "Piston Rings versus Cup Leathers" in hydraulic cylinders, we generally furnish piston ring construction. The majority of present-day hydraulic cylinder applications require relatively high speeds and frequent reversals, and many cylinders are used for reciprocating action. For this severe service, we have found piston rings most suitable, and with the use of oil as the hydraulic fluid medium, extremely long life is obtained. The success of piston rings depends on the fit of the rings in the piston and concentricity of the bore throughout the length of the cylinder. The latter requires honing of the cylinder bores to obtain a fine finish for the most satisfactory results.

However, piston ring construction is not advocated for every operation. Where the pressure is to be held for a period of time without movement of the piston, it is advisable to use cup leathers. Also where a slight out of roundness is present, we believe it advisable to use leather cup packing, except where service temperatures are too high for the efficient operation of cup leathers.

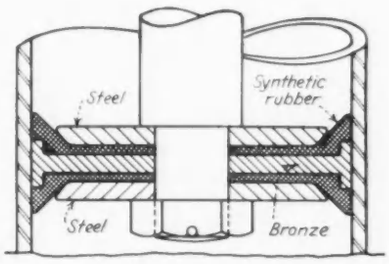
—OTTO J. MAHA
Chief Engineer
Hannifin Manufacturing Company

To the Editor:

●We use neither piston rings nor cup leathers in hydraulic cylinders for fire-fighting equipment, but a cup made of a rubber composition, which to date has proved very successful. To determine how this rubber was standing up we recently removed a piston from a cylinder that had been in service about four years. The rubber was found to be a trifle softer and the cup somewhat larger than it was when originally installed, but this did not affect its holding qualities. During service of four years the aerial ladder of this equipment had been raised and lowered some 5,000 times, according to the firemen who used it.

As shown in the illustration, two rubber cups are assembled back to back, so that the piston may be locked both bottom and top for holding the ladder in raising or lowering directions and so that the ladder may be used as a water tower. The center portion between the two rubber cups in made of bronze and the plates holding the rubber cups in place are made of steel, the cylinders being ground to size.

To illustrate the holding qualities of these cups, there are many times, while the truck is standing in the plant and being worked on, that the 100-ft. ladder is raised out of the bed about 2 ft. and has been held in this position for several days without any noticeable settling. Many times the ladder has been up within 6 in. of the building with several men climbing the ladder, holding this position for hours without any noticeable movement in the piston. From our experience, no liquid leaks past these rubber cups.



Composition rubber cups used back to back in hydraulic cylinder to hold 100 ft. aerial ladder in raised position without movement

Because of the various parts of the country in which these aerials are used, Prestone is used as a liquid, as we found it to withstand great velocities without gassing. We have tried several oils and other liquids and find the Prestone works the best.

H. P. LUNDSKOW
Chief Engineer
Peter Pirsch & Sons Company

Gear Pump Formulas

To the Editor:

●In the Intimate Correspondence columns of the April issue of *Product Engineering* J. P. L. of Philadelphia asks for a formula or method for calculating the displacement in gear tooth pumps for precision metering. A satisfactory formula for this purpose is not possible for general usage because the displacement depends not only on the tooth contour and its path in space but is also affected by change of shape of the pump casing traversed by the tooth. Each problem must be calculated from the completed geometric design using well known laws of mensuration because there is no standard design. Each designer uses mechanical movements to suit his particular needs. When a design is completed the displacement for another size may be calculated by proportionality if such proportionality has not been destroyed by mechanical necessity for strength of parts or for production reasons.

For pumps serving as metering devices, gear trains are used to translate the actual displacement to output as counted by the cyclometer. For this reason extreme accuracy of displacement is not necessary. The slip is a variable factor of more importance. In metering devices the shapes of tooth and housing are chosen to reduce fluid slip. The mass of the rotating members is kept small for sensitivity. The pressure drop across such metering devices must be small to keep slip low, using reasonable machining tolerances for production, and allowing a reasonable compensation for wear. The necessity for sensitivity and accuracy at reasonable cost usually necessitates a shunt operated device if large volumes are to be measured.

—HUGH E. WEIGHTMAN
DeKalb, Ill.

To the Editor:

● In the formula for gear pump volume displacements given by G. E. Breeze in the May issue, the second squared term within the brackets should be multiplied by the factor π, as is the first term. In this instance, the formula may be greatly simplified by performing the operations indicated in the brackets, assuming the form, $V = (2\pi WN)/P^2$, where V is the volumetric displacement per revolution in cu.in.; W, width of face in in.; N, the number of teeth in one gear; and P the diametral pitch (not the pitch diameter) of the gears.

In this simple method for calculating volumetric displacements it is assumed that the teeth of the gears are of involute profile with standard addenda. The equation should not be used for precision work as it gives values somewhat smaller than actual values.

—J. J. PESQUEIRA
New York, N. Y.

To the Editor:

●My critic is correct. The gear pump formula was incorrectly copied from my notes in too much haste, for which I most humbly beg your pardon.

—G. E. BREEZE
Highland Park, Mich.

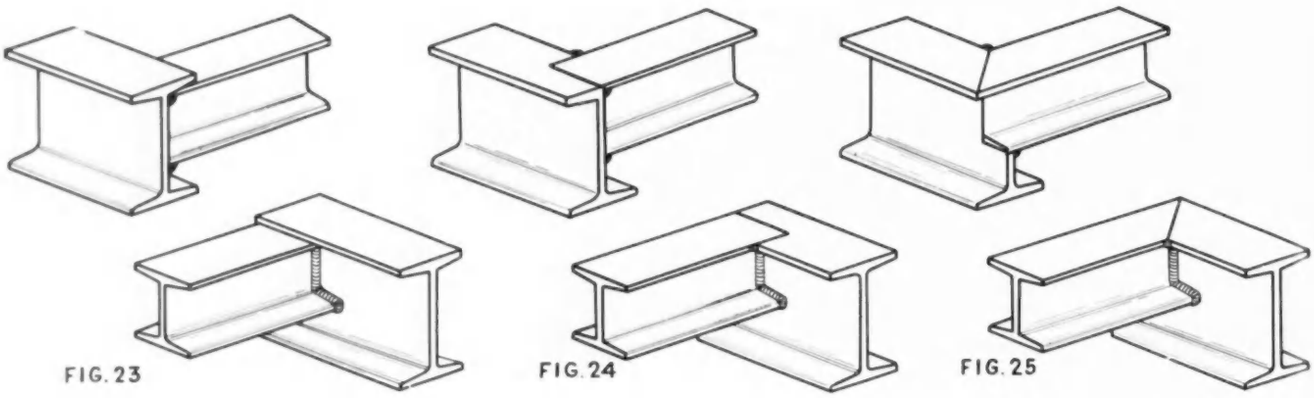


FIG. 23 FIG. 24 FIG. 25

Typical Joints of Arc-

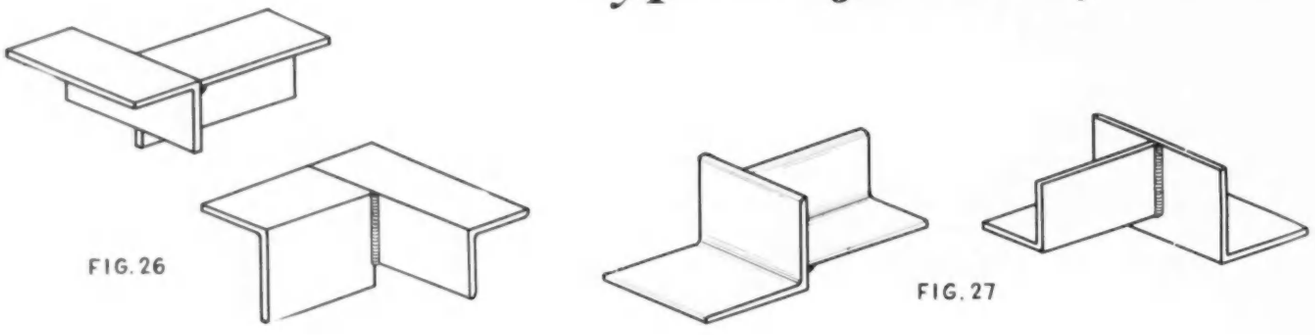


FIG. 26 FIG. 27

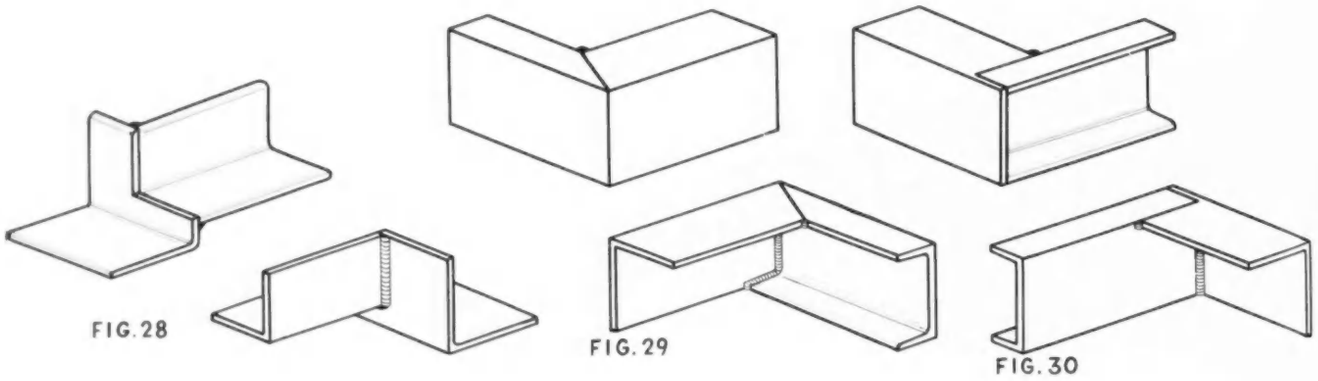


FIG. 28 FIG. 29 FIG. 30

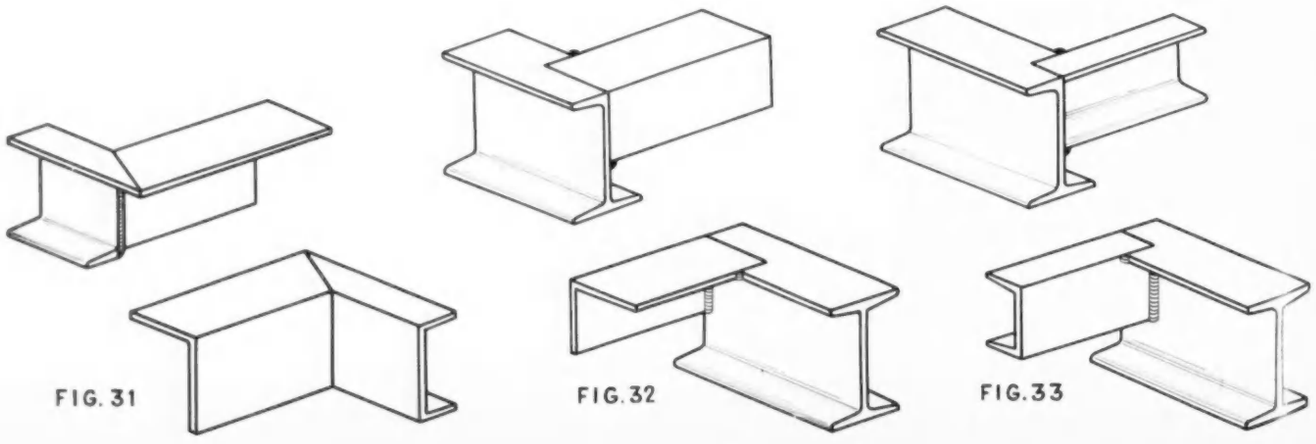


FIG. 31 FIG. 32 FIG. 33

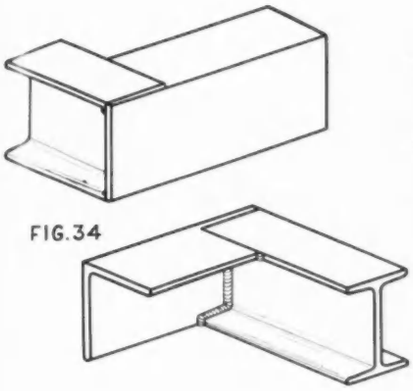


FIG. 34

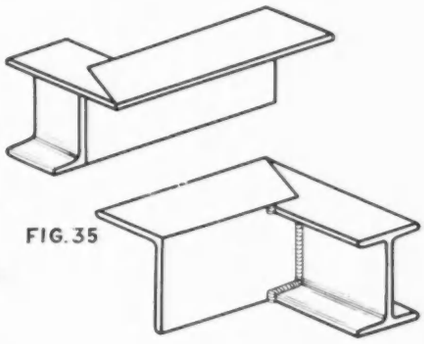


FIG. 35

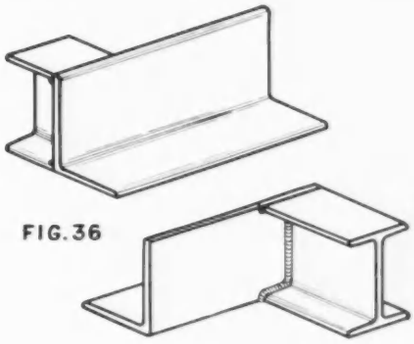


FIG. 36

Welded Structurals—II

R. A. GAST
The Lincoln Electric Company

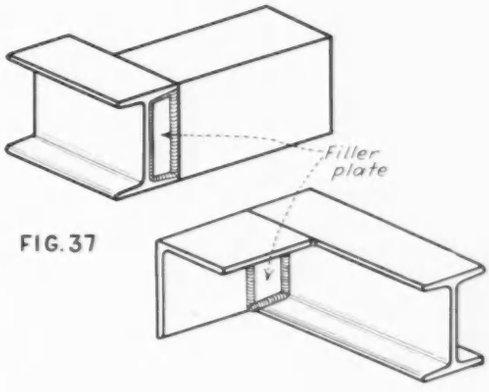


FIG. 37

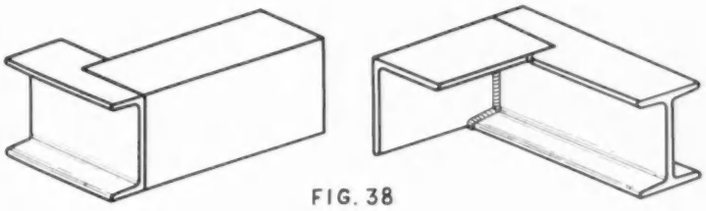


FIG. 38

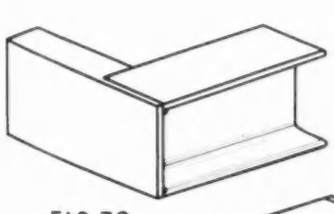


FIG. 39

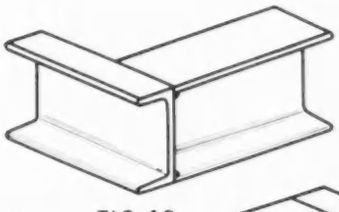


FIG. 40

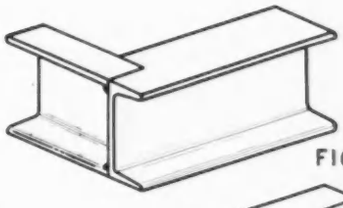
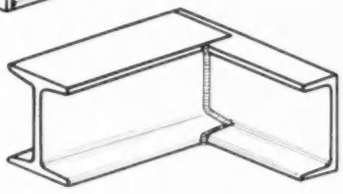
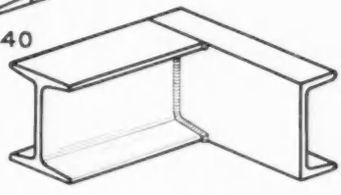
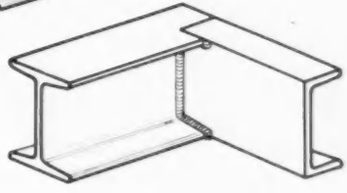


FIG. 41







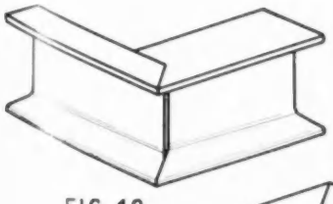


FIG. 42

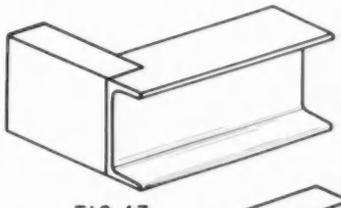


FIG. 43

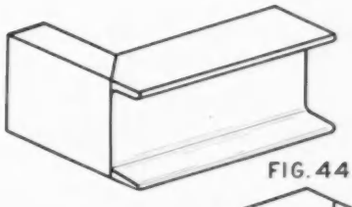
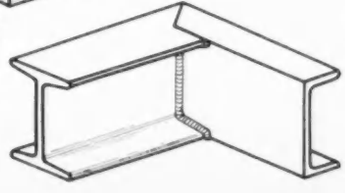
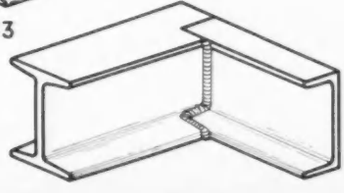
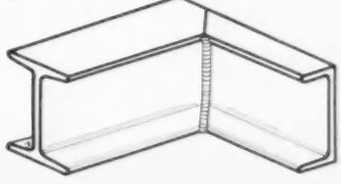


FIG. 44







NEWS

National Metal Show Expected to Break All Records

WHAT promises to be the greatest of all annual meetings ever held—the National Metal Congress and Exposition—will open in Cleveland on Oct. 19 and continue through to Oct. 23. The National Metals Exposition will be in the Great Cleveland Auditorium, while the Technical Meeting and Lectures will be held in the various hotels.

A complete program of technical sessions to be held by the American Welding Society in the Hotel Cleveland follows:

Monday, 2 p.m., Oct. 19

A. E. Gibson, chairman, The Wellman Engineering Company; E. Vom Steeg, vice-chairman, General Electric Company.

Welcome address by local official.

"Fundamentals of Metallurgy of Welding," by E. S. Davenport, and Dr. R. H. Aborn, United States Steel Corporation.

"Multi-Layer Oxyacetylene Pipe Welding," by R. M. Rooke, F. C. Saacke and A. N. Kugler, Air Reduction Sales Company.

High-Speed motion pictures of various welding processes, by E. Vom Steeg, General Electric Company; W. E. Crawford and Walter Richter, A. O. Smith Corporation.

Tuesday, 9.45 a.m.—Fundamental Research in Welding.

H. M. Boylston, chairman, Case School of Applied Science; E. R. Fish, vice-chairman, The Hartford Steam Boiler Inspection & Insurance Company.

"Heating by the Proximity Effect," by Edward Bennett, The University of Wisconsin.

"The Welding of Copper," by A. P. Young, Michigan College of Mining and Technology.

"Non-Destructive Testing of Welds," by W. B. Kouwenhoven, The Johns Hopkins University.

"Impact Tests of Welds at Low Temperatures," by Otto Henry, Brooklyn Polytechnic Institute.

Tuesday, 2 p.m.—Fundamental Research in Welding.

H. W. Gillett, chairman, Battelle Memorial Institute; R. E. Kinkead, vice-chairman, Consulting Engineer.

"X-Ray Methods for Studying Stress Relief," by John T. Norton, Massachusetts Institute of Technology.

pers to be presented.]

"Welded Beam-Column Connections," by Inge Lyse, Fritz Engineering Laboratory, Lehigh University.

"Circuit Characteristics and Arc Stability," by S. C. Osborne, Wilson Welder and Metals Co., Inc.

"Welded Structural Brackets," by C. D. Jensen, Lehigh University.

Wednesday, 9.45 a.m., Oct. 21—Technical Session.

A. F. Davis, chairman, The Lincoln Electric Company; C. A. McCune, vice-chairman, Magnaflux Corporation.

"Brazing with Silver Solders," by Robert H. Leach, Handy & Harman.

"Importance of Design Control for Welded Piping Systems," by T. W. Greene, Linde Air Products Company.

"Principles of Surfacing by Welding," by E. W. P. Smith, The Lincoln Electric Company.

"Technique for Resistance Welding Ferrous and Non-Ferrous Sheet Metals," by E. I. Larsen, P. R. Mallory Company.

Wednesday, 9.45 a. m., Oct. 21—

Hugh H. Dyar, chairman, The Linde Air Products Company.

"Procedures for Control of Welding Parts," by G. H. Moore, Jr., Newport News Shipbuilding and Dry Dock Company.

"Welding Copper and Its Alloys—A Review of the Literature," by Ira T. Hook, American Brass Company.

"Resistance Welding of Dissimilar Metals," by R. T. Gillette, General Electric Company.

"Thermit Welding," by J. H. Deppeler, Metal & Thermit Corporation.

"The Exploration of the Modern Metallic Arc," by L. J. Larson, A. O. Smith Corporation.

Thursday, 9:45 a.m., Oct. 22—Joint Session of American Welding Society with American Society of Mechanical Engineers.

C. W. Obert, chairman, Union Carbide & Carbon Research Labs., Inc.

"Stress Analysis," by C. H. Jennings, Westinghouse Electric & Manufacturing Company.

"Alloy Steels and Their Weldability," by A. B. Kinzel, Union Carbide & Carbon Research Laboratories.

"Welding of Alloy Steels," by W. L. Warner, Watertown Arsenal.

Thursday, 2 p.m.—Joint Session American Welding Society with American Society of Mechanical Engineers

H. F. Henriques, chairman, Air Reduction Sales Company.

"Welding Heavy Machinery and Equipment," by C. A. Wills and F. I. Lindemuth, Wm. B. Pollock Company.

"Modern Resistance Welding Developments," by A. E. Hackett, Thomson-Gibb Electric Welding Co.

"Using Steel Plates for Machine Frames." H. G. Marsh, Carnegie Steel Co.

In addition, the American Society of Mechanical Engineers will have two sessions on Friday, Oct. 23, one in the morning and one in the afternoon. The morning session will be devoted to the papers on the "Weldability of Non-Ferrous Metals" and the afternoon papers will be devoted to a "Review of Welding Developments as they affect Mechanical Design," the "Welding of Light Machines and Products," and the "Principles Involved in Selecting Casting versus Welding."

In addition to this comprehensive program, the American Society of Metals will have various sessions at which a total of 57 lectures dealing with every important phase of metallurgical engineering will be presented. According to Robert F. Archer, president of the American Society for Metals and chief metallurgist in the Chicago District for Republic Steel Corporation:

"This program gives the Society the largest and most comprehensive list of talks, ever scheduled, exceeding even last year's technical sessions. Papers will be presented each morning and afternoon of the convention, with the exception of Wednesday morning, when the society's annual meeting will be held and the 1936 Edward DeMille Campbell Memorial Lecture will be presented by J. P. Gill, chief metallurgist of the Vanadium Alloys Steel Company."

Although on account of the great number of papers to be presented it will be necessary to stage simultaneous sessions, these have been arranged so that there will be no conflict with regard to subjects. Papers on the subject will not be presented at the same time, and hence the members will be able to hear all important lectures relating to their particular branch of the metal industry. Features on the technical program include lectures on "Physical Testing of Metals," by H. D. Churchill, Professor of Mechanics at Case School of Applied Science, and three lectures on X-ray by Dr. Kent R. Van Horn, metallurgist at the Aluminum Company of America.

[EDITOR'S NOTE—October *Product Engineering* will contain the complete final program for the whole congress and also abstracts of many of the pa-

New Acid Tests for Fired Enamels

IN COOPERATION with the Porcelain Enamel Institute, the Bureau of Standards has been engaged in developing a test by which the acid resistance of vitreous or porcelain enamel can be determined. The problem has been primarily one of developing a test by which the degree of resistance of the different enamels can be compared by a test which will merit adoption as a standard. Heretofore, one method of testing has been to determine the heat concentration of a given acid which would cause visible attack under specified conditions. Seven concentrations of citric acid were used, the strongest being 64 times the concentration of the weakest.

Another test method was that of determining the degree of attack caused by a specified treatment with acid. This method gave promising results. Three degrees of attack could be distinguished visually: namely, (a) no visible attack, (b) attack visible, but insufficient to decrease ease of erasure of a pencil mark, (c) attack sufficient to prevent erasure of a pencil mark with a wet cloth.

In the test methods being developed by the Bureau of Standards, the loss of specular (mirror) reflection caused by the acid, as determined by a Hunter glossmeter, was taken as a criterion of acid resistance. The principal advantage of this method is that it makes possible the detection of numerous degrees of acid resistance by comparing the original value of the specular reflection index with its value after treatment of the test specimen with acid.

Paris Colors Now Cabled to N. Y.

ON Wednesday, Aug. 12, the first commercial trans-Atlantic color dispatches were cabled from Paris to Marshall-Field & Company in Chicago. Immediately upon receipt of the "cabled colors" production on exact duplicates of the original Paris colors was begun and within 48 hours the true reproduction of original Paris fashion creations were placed on sale in department stores.

The only equipment required for "color cable," as the service is termed commercially, is a machine about the size of a large Gladstone bag for sending, and a corresponding machine on the receiving end. In the cabinet of the transmitting machine are color corrected lights, incandescent lamps to demonstrate the appearance of color samples under ordinary indoor illumination, and a calibrated wheel driven by a universal motor. Delicate volt meters assure precise duplication of the different currents in use. Completing the necessary color cable equipment is a special set of 87 Maxwell color disks.

At the transmitting end, the operator selects a number of variously colored disks and inserts them on the calibrated electrically-driven wheel. Whirling the wheel merges the disks into one solid color. The surface percentages of the disks are added or subtracted until the color appearing on the spinning wheel precisely matches the sample. Percentage reading of the exposed surfaces of the various disks are then taken and the number and color designation of the disks are telegraphed or cabled to the receiver. On the receiving end, the disks are placed on the calibrated wheel of a second color cable machine and set to the percentages which conform with the specifications contained in the cabled message. When the machine is set in motion, an infallible color match results. A typical color cable would read "Sixteen mustard yellow ninety-three stop fifty-six point five degrees twenty-six stop forty-three point five degrees."

By using this new system, any one of 300,000 color variations can be accurately expressed in numerical terms which in turn can be cabled or telegraphed to any point in the world where there is a telegraph or telephone.

MEETINGS

American Gear Manufacturers Association—Semi-annual convention, Sept. 8–10. To be held on the S. S. Seeandbee sailing from Chicago on Tuesday morning Sept. 8 and docking at Cleveland, Sept. 10. J. C. McQuiston, Manager-Secretary, Penn Lincoln Hotel, Wilkensburg, Pa.

International Acetylene Association—37th Annual convention. Nov. 18–20, Jefferson Hotel, St. Louis, Mo. International Acetylene Association, 30 East 42 St., New York, N. Y.

EXHIBITIONS

Great Lakes Exposition—Cleveland, Ohio, June 29–Oct. 4. Lincoln G. Dickey, manager, Terminal Tower, Cleveland, Ohio.

National Electrical and Radio Exposition—Madison Square Garden, Sept. 18–28. Joseph Bernhard, managing director, Grand Central Palace, New York, N. Y.

National Business Show—Exhibition of office equipment, Commerce Hall, Port Authority Building, New York. Oct. 19–24. Frank E. Tupper, 50 Church St., New York, N. Y.

National Metals Congress & Exposition—1936 Annual meeting, Cleveland, Ohio, Oct. 19–23. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio. The following national societies will participate: American Welding Society, the Wire Association, the Institute of Metals, the American Society for Metals, the American Institute of Mining & Metallurgical Engineers, and the American Society of Mechanical Engineers.

Merger Announced

ON AUG. 5 was announced the merging of the business of the Unyte Corporation with that of the Plaskon Company, Inc. The consolidated organization to be known as the Plaskon Company, Inc. It is claimed that this new alignment, the first consolidation in the plastics industry for many years, will be by far the world's largest producer of urea formaldehyde resins. The new concern takes over all the processes and patent rights formerly held by the Unyte Corporation which made urea plastics, essentially the same products as those manufactured by Plaskon.

Do You Know That——

• Tungsten can now be electrodeposited in a homogeneous combination with one or more metals through the same electrolytic solution. Some of the various combinations are tungsten-nickel, tungsten-cobalt, tungsten-iron, tungsten-molybdenum, tungsten-tantalum and also combinations such as tungsten-nickel-cobalt, tungsten-nickel-iron, tungsten-nickel-molybdenum, and numerous other combinations. Even quaternary combinations, such as tungsten-nickel-cobalt-iron can be electrodeposited. Electro-platings possessing unusual properties are thus achieved. (*27*)

• A recently developed finish is claimed to be heat-resisting to the extent that it can be used for the top of stoves, or temperatures up to 700 deg. F. (*28*)

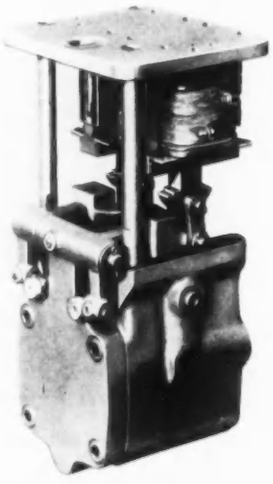
• Cotton fabric used in making fire hose is now being treated with rare earth metal salts having radio-active properties. It is said that fabric thus treated will effectively and permanently resist mildew and make the hose moisture repellant, rot and freeze-proof. (*29*)

(For details, write the editor, giving the item number.)

New Materials and Parts

Remote Controller

Combines hydraulic and electric devices to obtain required functions or cycles of fluid power feeds, no mechanical linkage is used. Controller is mounted on feed pump. Energizing and deenergizing two solenoids intermittently or in unison through remote push-button station or limit switches operates and controls the movement of the pilot valves and selects the flow of oil to the control cylinder and for the function required. The controller unit can be mounted on the right or left hand side of pump manifold. Styles are available with provisions for four and five feed functions. Oilgear Co., Milwaukee, Wis.



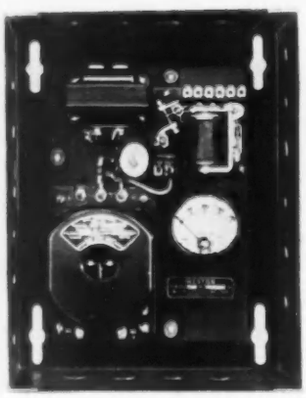
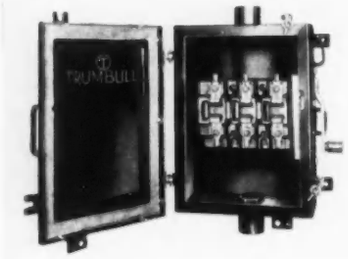


Photo-Cell Control

Applicable on industrial installations where the interruption of a light beam provides the initial impulse for opening or closing an electrical circuit. Consists of a photo-cell in a protective mounting, a relay unit in a separate panel box, and a light source. A light intensity of 100 foot-candles on the cell without external amplyfing tubes, will operate relay system with output capacity of 500 watts. The relay is arranged for single-pole double-throw operation, with an overall speed of response of 0.15 seconds, and operating speeds of 400 per min. A microammeter calibrated in light units shows the level of illumination reaching the photo-cell. Weston Electrical Instrument Corp., Newark, N. J.

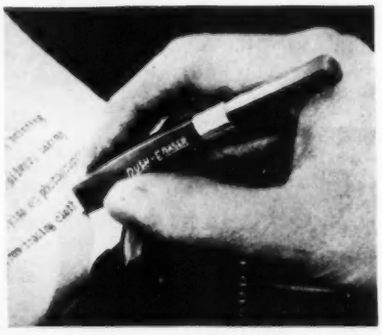


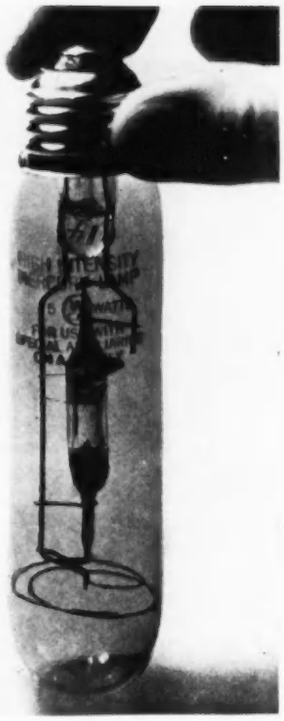
Safety Switch

For hazardous locations, classified by Underwriters as Class II, Groups "F" and "G", approved for use in atmospheres containing dusts of metal, coal, coke, or grain. Claimed to be dust tight, water and weather proof. Furnished in heavy gage steel boxes with gasketed covers fastened by wing nuts. Available in 3 pole types, no fuse, 30 to 200 amp. inclusive. Trumbull Electric Mfg. Co., Plainville, Conn.

Brush-Eraser

Brush consisting of spun glass fibers treated with a flexible binding solution. Removes ink, typewriting, india ink from tracing cloth, and cleans and polishes metal surfaces. In a holder which fits the hand, the top section of which turns to propel the brush through the tip as needed until used up. Lower half of the holder unscrews to permit the insertion of a refill. The Eraser Co., Inc., 120 E. Washington St., Syracuse, N. Y.



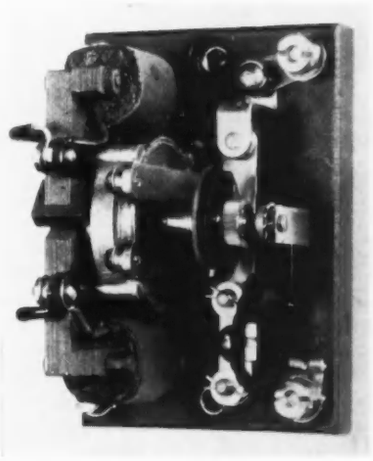


Midget Mercury Lamp

Especially suitable where smoky or dusty industrial conditions prevail. The strong blue and green rays in this light may encourage applications in photographic work. The light-giving element is a small mercury vapor filled bulb of fused quartz. An outer bulb of glass serves as a heat insulator and also protects the inner supporting structure. Must be operated in conjunction with a transformer. Available in two types, one for burning base up and one for burning in horizontal position. Capacity 85-watt, overall length 5⅝ in., and the base is medium screw. Westinghouse Lamp Co., Bloomfield, N. J.

Timing Devices

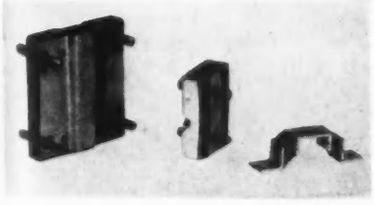
Motor operated devices of the thermal, inertia, air dashpot and capacitor types. One is of the immediately recycling type in which a synchronous



motor drives cams through a chain of gears. The timing is adjustable over a wide range and practically any arrangement of load contacts can be furnished. Another is of the continuously rotating type which by means of gears the cams of the timer are made to revolve at 1 r.p.m. or any other desired speed. The contacts can be arranged to be closed for any time up to half the time required for one revolution if the timing is adjustable, or up to the full revolution time if the timing is not adjustable. Struthers Dunn, Inc., 139 N. Juniper St., Philadelphia, Pa.

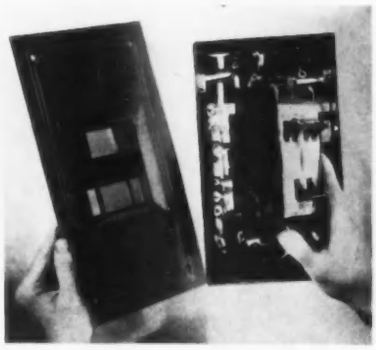
Vibro-Insulators

Mountings are of the shear type consisting of an age-resisting rubber attached to two metal plates with a brass plating type of bond which gives an adhesion of 200 lb. per sq. in. Designed for bases of machines, electric and combustion motors, generators, vibrating screens and reciprocating mechanisms. Available in three types. Capacities are; for Type No. 1, 800 lb., Type No. 2, 200 lb., and for Type No. 10, 50 to 3,000 lb. B. F. Goodrich Co., Akron, Ohio.



Multi-Breaker Load Center

Consists of one or more automatic circuit breakers grouped in a sealed molded housing. Applicable for residential service for control and protection of circuits for domestic appliances, ranges, and water heaters. The mechanism of each circuit breaker includes



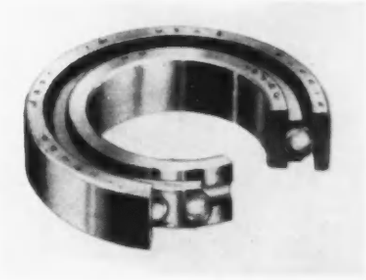
a conducting bar resting on a fulcrum, one end carrying the moving contact and the other engaging the bimetallic tripping element which is brazed to the branch circuit terminal. A stationary contact is common to all the multi-breaker poles. The bar is rocked and actuated by a cam shaped portion of the operating handle. The contacts are of a non-welding silver alloy. All terminals are of the solderless type. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., and the Square D Co., Detroit, Mich.

Sensitive Solenoid Relay

Designed especially for operation with a one tube amplifier operated by photocells. Operates entirely by solenoid action and gravity, has a top adjustable pole piece and lock nut for adjusting to the desired "drop out" or "pull in" value of energizing current. Is in the form of a cylinder $1\frac{3}{16}$ in. diameter by $2\frac{5}{16}$ in. long, overall length is 3 in. including top screw and Octal base. The contact arrangement is single-pole double-throw, with double break on both throws and separate isolated contactors. Contacts and contactors are of fine silver and will handle 100 watts on each circuit. Available in bright cadmium finish or black crinkle enamel. Acusto-Lite Laboratories, 2920 S. Vermont Ave., Los Angeles, Calif.

Mazic Zinc Plating

Process permits heavy deposits from a cyanide solution with a uniformly bright protective coating said to be cheaper and comparable to cadmium in appearance and durability. Is a high-speed plating bath with the cathode current density much higher than in ordinary zinc solution. Tanks should be of unlined steel. Satisfactory results can be obtained in still tank, semi- and full automatic plating operations. Hanson-Van Winkle-Munning Co., Matawan, N. J.



Compact Ball Bearings

Designed specifically for machine tools where unusual compactness is required. Provides standard ball bearings for applications formerly requiring bearings of special design, for anti-friction mounting of closely spaced spindles on high-speed grinders and boring machines. Because of the enlarged bores which accompany standard outside diameters, larger centers, adapters, and arbors can be used to increase spindle rigidity without an increase in overall dimensions. Are of the "super-precision" type in which all tolerances are held within particularly narrow limits. Load capacities are said to be exceptionally high for bearings of these dimensions. Designated as the 9500 Series, bearings are available in bore sizes ranging from 20 mm. to 260 mm. Sizes up to 100 mm. bore are equipped with solid composition retainers, on the larger sizes machined bronze retainers are used. Fafnir Bearing Co., New Britain, Conn.

Impregnating Resins

Several improved thermosetting resin solutions for the impregnation and coating of motor and generator armatures. These resins set up to a permanent and infusible film by chemical reaction. Other properties claimed are higher insulation and

greater resistance to water, oils and chemicals, and a stronger mechanical bond between wires and armature frames. Metal castings are also impregnated to reduce porosity, subsequent baking hardens the resin infusibly. General Plastics, Inc., North Tonawanda, N. Y.

Pneumatic-type Controller

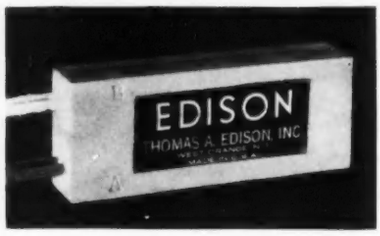
Developed for automatic temperature and pressure control installations that do not warrant a chart record. Known as Model 90, the measuring element is the same type as employed in the recording instruments, and is equipped with a wide-range sensitivity adjustment. Can be changed from direct to reverse acting, or vice versa, without additional parts. The case is moisture-, fume-, and dust-proof. Available for wall and for flush panel mounting. The Bristol Co., Waterbury, Conn.

Self-Cleansing Filter

Consists of a motor-driven pump which draws the coolant through the filtering element and pump housing. Designed primarily for machine shop use on grinders and applications where a continuous flow of clean liquid must be maintained. The coolant passing through the housing strikes against vanes that rotate the filtering element. A part of the coolant is diverted and forced under pressure into a vertical tube with a narrow slot extending its length. Rotation of the element brings the surface opposite the slot to clean the filter. Vertical fins on the inside prevent the coolant from swirling. Available in

two capacities of 18 gal. and 30 gal. per min. A ¼ hp. motor drives the small filter and a ⅓ hp. motor the larger one. Motor Improvements, Inc., 365 Frelinghuysen Ave., Newark, N. J.

Time Delay Relay



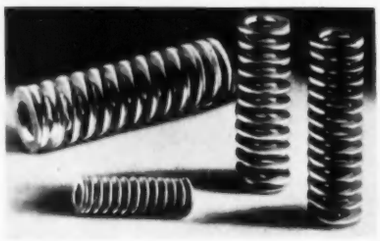
Operates as a thermal inertia device depending for its period upon the time required to heat the block to a predetermined temperature. Is applicable for timing alarm bells or lights, also to heating devices for removing from the circuit an accelerating heat-up unit as operating temperature is attained. Compensation for ambient temperature and changes in line voltage is automatic. Consists of a small metal block approximately ⅝ in. x 1½ in. x 2½ in. into which the control element and a small electrical heater are assembled. Thomas A. Edison, Inc., Edison Electrical Controls Division, West Orange, N. J.

Anti-Corrosive Enamels

Made of rust inhibiting pigments and a synthetic resin vehicle, to form a hard, tough coating resistant to wear and of pleasing appearance. Claimed to be effective protection from the corrosive action of humidity, salt water, alkali, and a variety of chemicals, and in preventing electrolytic action. Can be applied by either brush or spray and air-dry in four hours. Types for baking for one hour at 250 to 350 deg. F. can be supplied. Available colors are brown, black, red, and a bronze liquid, which serves as a vehicle for aluminum and other metallic powders. Maas & Waldstein Co., Newark, N. J.

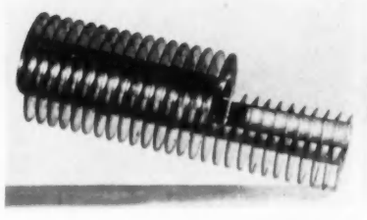
Die Springs

Flat rounded type cross-section. Coiled from special alloy keystone section wire, which when coiled assumes a rectangular cross section to give it equal section and equal strength on both inside and outside spring diameters. Available in 15 standard free lengths for hole diameters ¾ to 2 in. by ¼ in. increments. It is claimed that these springs are engineered with stresses to give high load capacity, long life, and flexibility. Danly Machine Specialties, Inc., Chicago, Ill.



Heat-Exchanger Tube

Formed from a heavy wall tube. Fins are thrown up on the surface of the tube in the form of a helix with no joint at the tube. The tapered shape of the fin with thin outer-edge and heavy cross-section adjacent to the tube gives maximum heat dissipation for a minimum of material. When maximum bending properties are desired, it is made with a corrugated inside surface so that the outside radius can readily elongate and the inside compress. When the ends of the tube are to be stripped for rolling into headers, a smooth inside surface is furnished. Available in either copper or aluminum in nominal diameters of ⅜ and ⅞



in. with 9 fins per inch with a fin height of 3/32 in., in tube diameters ½ to 1 in. inclusive with 4, 6 and 7 fins per inch with a fin height up to ⅛ in. and in 3 and 4 in. tube sizes with 4 fins per in. ⅛ in. high. Wolverine Tube Co., Detroit, Mich.

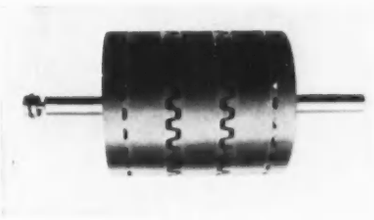
Incandescent Lamp Blueprinter

Double the size of previous models with all essential features retained, including the use of incandescent lamps as a light source and elimination of the belt feed by the flat bed printing surface. Has a timing switch of the heavy duty type for automatically breaking the heavier current. Forced

ventilation is supplied to prevent any tendency to overheat. Size, 2 ft. 4 in. by 3 feet. Milligan & Wright Co., 1237 W. Third St., Cleveland, Ohio.

Magnetic Pulley

Unit for separating process installations. The conveyor belt forces air through corrugated ducts in the pulley to remove heat generated in the coils. Bronze spacer rings bridge the gap between the poles. Furnished with



turned and polished shaft, a set of collars, bronze slip rings, double contact brushes, dust-proof collector ring housing, switch and cabinet, and a resistance to absorb the counter emf when circuit is opened. Dings Magnetic Separator Co., Milwaukee, Wis.

Resin for Lacquers

For use especially in cellulose ester lacquers of high gloss, high solid content type. Suited for automobile finishing and refinishing, refrigerators, kitchen furniture, cabinets, bathroom fixtures and imitation tile hospital furniture, textile and paper coatings, metal lacquers for hardware and instruments. In combination with nitrocellulose are said to yield lacquers of exceptional flow and gloss, and to be unusually resistant to moisture, oxidation, strong soaps, acids, oils and greases. Their solutions in common lacquer solvents are low in viscosity, permitting the formulation of high solid content lacquers with film building properties. Bakelite Corporation, New York, N. Y.

MANUFACTURERS' PUBLICATIONS

Ampco Metal—Ampco Metal Inc., Milwaukee, Wis. Catalog No. 21. 32 pages, 8½x11 in. Information covering plant and metals data, covering general uses and grade recommendations. Physical properties and applications of various grades of alloys are described. Ten pages are devoted to centrifugal castings, performance in acid service, metal dies, bushings, bearings, and gears. Notes are included for machining Ampco metal, with data for patterns, welding, tinning, brazing and heat-treatment.

Circular Slide Rule—Wrought Washer Mfg. Co., 2100 S. Bay St., Milwaukee, Wis. Slide rule, 8 in. in diameter. Calculates the weight per thousand pieces, as well as pieces per pound of any size washer made of steel, brass, copper, aluminum, fiber, paper, rubber cork, celluloid or wood.

Colorimeter Stand and Daylight Illuminator—C. J. Tagliabue Mfg. Co. Park and Nostrand Aves., Brooklyn, N. Y. Bulletin No. 1143, 1 page, 8½x11 in. Describes new adjustable stand used to support and illuminate either the Tag-Saybolt chromometer, or the Tag-Robinson colorimeter, using as a light source the standard Tag-Macbeth daylight lamp with diffusion screen.

Controlled Rectifier—Ward Leonard Electric Co., Mt. Vernon, N. Y. Bulletin 8851, single page, 8 x 10½ in. Describes apparatus for providing the "Ward Leonard System" of control for a d.c. motor from an a.c. power supply, separately exciting the motor field by means of one part of the rectifier and supplying armature current at a predetermined and adjustable potential from another section of the controlled rectifier.

Fractional Horsepower Motors—Bodine Electric Co., 2254 W. Ohio St., Chicago, Ill. Bulletin 1020, 4 pages, 8 x 10½ in. Outlines a line of split-phase, capacitor, shaded pole, polyphase and synchronous a.c. motors, also shunt and compound d.c. motors. Contains data on constant speed governor motors, and standard and heavy-duty speed reducer motors.

Electronic Tubes—Westinghouse Lamp Co., Bloomfield, N. J. 5 descriptive data sheets, 8½ x 11 in. Describe ratings and operating characteristics of amplifier and oscillating tubes; grid glow tubes and ignitrons; phototubes; rectifier tubes; and miscellaneous types for industrial applications.

Heavy-Duty D.C. Motors—Reliance Electric & Engineering Co., Ivanhoe Road, Cleveland, Ohio. Bulletin 8½x 11 in. Bulletin 213, 2 pages, and Bulletin 214, 4 pages, describe advantages and constructional features of fully-inclosed fan-cooled d.c. motors. Bulletins 215 and 216, each 4 pages, describe features of 6 pole heavy-duty d.c. motors.

Indicating and Control Instruments—Brown Instrument Co., Wayne & Roberts Aves., Philadelphia, Pa. Booklet, 24 pages, 7 x 10 in. Treats from the executive viewpoint, of the advantages gained by the use of instruments, with examples of industrial applications and illustrations.

Leather Belting Manual—E. F. Houghton & Co., 240 W. Somerset St., Philadelphia, Pa. Booklet, 40 pages, 8½ x 11 in. Contains transmission data, engineering tables, and belt formulas, with notes on the selection and care of leather belts, types of lacings and cementing instructions.

Magnetic Pulleys and Separators—Dings Magnetic Separator Co., Milwaukee, Wis., Catalog No. 25, 30 pages, 8½x11 in. Deals primarily with magnetic pulleys and pulley type separators with details and data. Five pages are devoted to illustrating and describing standard and special designs. Tables are given of capacities, dimensions, and weights of magnetic pulleys.

Manufacturing by Shielded Arc Welding—The Lincoln Electric Co., Cleveland, Ohio. Booklet 16 pages, 8½x11 in. Sets forth the advantages of shielded arc welding in manufacturing machinery, with illustrations and cuts of typical examples of welded parts and their application to machinery.

Meehanite Metal Chart—Meehanite Metal Corp., Pittsburgh, Pa. Chart, 17 x 15 in. Recommends grades of Meehanite metal to use for various service requirements, with graphs showing stress-strain relations, effect of heat-treatment, comparative wear, erosion, rate of corrosion, and relative growth after heating.

Nickel Alloy Steels—International Nickel Co., Inc., 67 Wall Street, New York, N. Y. Booklet, 20 pages, 8½x 11 in. Buyers' guide to warehouse stocks of nickel alloy steels, with S.A.E. numbers and sizes carried. Contains a list of trade names of the principal alloys with producer's name, also chemical composition of material. Four pages are devoted to the properties and applications of constructional nickel alloy steels.

Roller Chains and Sprockets—Link-Belt Co., 519 N. Holmes Ave., Indianapolis, Ind. Catalog No. 1557, 24 pages, 8½ x 11 in. Stock list of sprockets, roller chains for drives, and attachments. Contains horsepower ratings and speed ratio tables.

Oil Circuit Breakers—Roller-Smith Co., 233 Broadway, New York, N. Y. Catalog 6—Sections 2 and 4, each 2 pages, 8½x11 in. Section 2 covers face plates, attachments, and accessories for hand operated breakers. Section 4 describes small breakers with capacity ratings of 50,000 kva., and less.

Portable Cable—General Electric Co., Schenectady, N. Y. Bulletin GEA—1918A, 28 pages, 8 x 10½ in. Gives data and listings of tellurium-compounded all-rubber jacketed, glyptal base compound sheathed, and braided types of portable cable for mining, welding and transit equipment. Information on cable connectors, terminals and couplers is also included.

Power Transmission Appliances—T. B. Wood Sons Co., Chambersburg, Pa. Condensed catalog No. 179, 15 pages, 3¾x6½ in. Illustrates adjustable drop hangers, post hangers, pillow blocks, couplings, friction clutches, V-belt drives, and includes tables of standard sizes.

Silent Chain Drives—Ramsey Chain Co., Inc., Albany, N. Y. Catalog 636, 36 pages, 6¼x9¼ in. Story of the silent chain in industry, with many illustrations of applications, details of links and roller bearing chains, with notes on standard details. Engineering tables and design data with examples of chain drive are included with horsepower ratings. Installation instructions are also included.

Steel Data Book—Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill. Handbook, 192 pages, 4½ x 8½ in. Contains shop data on tool and alloy steels, covering heat-treatment, forging, annealing, quenching and drawing. Charts are shown for the average physical properties of heat-treated steels. Seven pages are included of definitions of terms common to the steel industry. Engineering data are listed covering miscellaneous weights and steel specifications.

Technical Coatings—Technical Coatings, Inc., 9–15 Park Place, New York, N. Y. Booklet, 10 pages, 8½x11 in. Contains descriptive matter recommending coatings used for protection against corrosion, including primers, topcoats, and colors. No illustrations are used.

Vari-Speed Motor Pulley—Reeves Pulley Co., Columbus, Ind. Catalog V-367. Ten pages, 8x11 in. Describes the advantages, operating principles and construction details of Vari-Speed motor drive pulleys. Tables of engineering information are given showing horsepower, pulley sizes, and belt lengths. Contains many illustrations showing various applications.

BOOKS AND BULLETINS

Welding Technology And Design

G. F. P. Fox and F. Bloor. 90 pages, 5¼x7½ in. 35 illustrations, 18 tables. Red clothboard covers. Published by J. B. Lippincott Co., Washington Square, Philadelphia, Pa. Price $1.50.

This volume presents descriptive material taken from British sources in a simple direct manner. Designers and draftsmen desiring a general knowledge of elemental design and fabricating procedure in welding, will appreciate the descriptions of methods, illustrations and data tables. Contents cover welding methods, types and properties of welded joints, weldability of the ferrous metals, physical testing, and electrodes. Also describes arc resistance, spot, and atomic welding machines. The last chapter deals with examples of welding in the structural, electrical, automobile, shipbuilding and general engineering fields.

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Durez Molding Materials

General Plastics, Inc., North Tonawanda, N. Y. 28 pages, 5½ x 9 in.

This publication is a products list in bulletin form. It gives in table arrangement physical property data on all of the standard Durez phenolic molding materials, including blacks and colors. Practical information is also presented on molding technique and part design. Molders, manufacturers having molding equipment, and those buying molded parts will find much of interest and help in its pages.

Tin Research

Report published by the International Tin Research Council. Copies are available free from L. J. Tavener, 149 Broadway, New York, N. Y.

"Opacifiers in Wet and Dry Enamels." Series A, No. 42, 32 pages.

American Lubricating Greases

J. I. Clower, Bulletin No. 35, Engineering Experiment Station Series. Virginia Polytechnic Institute, Blacksburg, Va., 80 pages, 6x9 in. Paper cover.

Information concerning the composition, manufacture, uses and testing of American greases is given in brief non-technical wording. The point of view of the user has been emphasized throughout rather than that of the manufacturer, so that those interested can obtain a clearer appreciation and a better understanding of greases and their application.

Beginning with a short review of some very early lubricating methods, the author describes the art, handicraft, and industrial science of grease making. The materials and equipment used in the cooked, cold sett, and residuum processes form the subject matter of the first chapter. Physical, chemical, and mechanical tests and the apparatus employed in each make up the larger part of the bulletin along with an outline of the factors involved, and the characteristic qualities required by the various specific services. The methods by which grease can be applied to bearings, gears, slides and mechanical parts are also classified.

Many illustrations and tables are used to supplement the text. A bibliography is appended, and inserted for handy reference is a chart stating the classifications, composition, and uses of greases.

Thermometers

The Bristol Co., Waterbury, Conn. 88 pages, 8x10½ in. Paper covers, punched for two ring binder.

This catalog, printed in the form of an engineering reference book, contains timely and useful information concerning the theory and practice of present-day thermometry.

It describes liquid-filled, vapor-tension, and gas-filled recording, indicating, and controlling thermometers. In connection with automatic temperature controllers, both electric and pneumatic types, sketches are included with information for application. Data are also given on industrial stem thermometers, resistance thermometers, humidity measuring instruments, control valves, time-temperature controllers and accessories for industrial applications. Over 500 temperature recording charts are illustrated in full size.

Mine Car Bearing Data

The Timken Roller Bearing Co., Canton, Ohio. 64 pages, 8½ x 11 in. An enlarged edition of the Mine Car and Mine Locomotive Section of their Engineering Journal.

This supplement presents a complete listing for all types and classifications of mine cars ranging in capacity from 1½ to 35 tons. The volume consists of plate drawings giving suggestions for the application of Timken Tapered Roller Bearings, covering various types and combinations of bearing mountings, closures and nuts. Data is also included on bearing selection, fitting practices, assembly, lubrication and maintenance.

Mining engineers will find this information helpful in making preliminary drawings, and in solving bearing problems that arise in connection with car design.

Alcoa Aluminum and Its Alloys

Aluminum Company of America, Pittsburgh, Pa. 92 pages, 5½ x 8¼ in. Paper covers.

Recognizing the rapidly growing demand of designing engineers for information concerning the properties of aluminum and its alloys, the Aluminum Company of America presents in this booklet some fundamental information concerning the alloys produced by them.

The subject matter of the first half of the book is devoted to a description of the qualities and properties of aluminum, wrought and casting alloys, methods of working and fabricating, resistance to corrosion, annealing practice, and theory of heat treatment, and is illustrated with pictures of typical applications.

Tables are included in the appendix showing the physical and mechanical properties, compositions, dimension tolerances, available commercial sizes and forms of the various aluminum alloys and aluminum alloy products.

Slider-Crank Mechanism

J. L. SWYERS

Atlas Imperial Diesel Engine Company

THE equation for the velocity v of the piston or crosshead in a slider-crank mechanism is

$$v = -r\frac{d\theta}{dt}(q_1 \sin\theta + 2q_2 \sin 2\theta + 4q_4 \sin 4\theta + 6q_6 \sin 6\theta \ldots)$$

In this equation r is the crank radius and θ is the crank angle measured as indicated in the accompanying figure. The values of q are a function of the ratio of the crank radius to the length of the connecting rod. (See p. 323, August *P.E.*) Since acceleration is the first derivative of the velocity, with respect to time, its equation is:

$$a = \frac{dv}{dt}$$

$$= -r\left(\frac{d\theta}{dt}\right)^2(q_1\cos\theta + 4q_2\cos 2\theta + 16q_4\cos 4\theta + 36q_6\cos 6\theta + \ldots\ldots\ldots\ldots)$$

$$= -r\omega^2 F_3$$

wherein ω = velocity in radians per sec.

$$F_3 = (q_1\cos\theta + 4q_2\cos 2\theta + 16q_4\cos 4\theta + 36q_6\cos 6\theta + \ldots\ldots)$$

The table below gives the value of F_3 for different crank positions and ratios of length of rod to crank radius.

If n is the r.p.m. then

$$\omega = \frac{2\pi n}{60}$$

and $$a = F_3 r\left(\frac{2\pi n}{60}\right)^2$$

Thus the acceleration can be determined readily by selecting the proper value of F_3 from the table If r is in in., a will be in. per sec. per sec., while r in ft. gives a in ft. per sec. per sec.

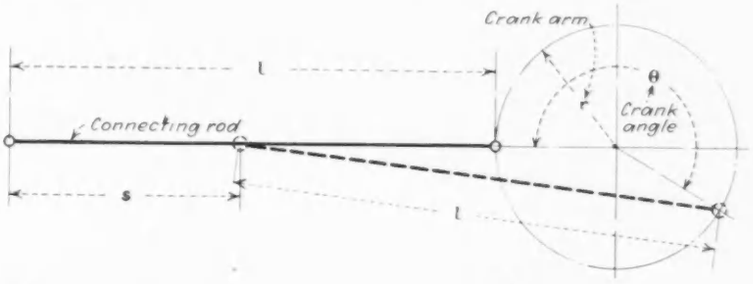

Table III—Factors F_3 For Determining Piston Acceleration

Crank Angle Degrees	RATIO OF LENGTH OF ROD TO LENGTH OF CRANK															
	3.5	3.6	3.7	3.8	3.9	4.0	4.1	4.2	4.3	4.4	4.5	4.6	4.7	4.8	4.9	5.0
0–360	1.286	1.278	1.270	1.263	1.256	1.250	1.244	1.238	1.233	1.227	1.222	1.217	1.213	1.208	1.204	1.200
5–355	1.278	1.270	1.263	1.256	1.249	1.243	1.237	1.231	1.225	1.220	1.215	1.210	1.206	1.202	1.197	1.193
10–350	1.254	1.247	1.240	1.233	1.226	1.220	1.215	1.209	1.204	1.199	1.194	1.190	1.185	1.181	1.177	1.173
15–345	1.216	1.208	1.202	1.196	1.190	1.184	1.179	1.173	1.169	1.164	1.159	1.155	1.151	1.147	1.143	1.140
20–340	1.162	1.156	1.150	1.144	1.139	1.134	1.129	1.124	1.120	1.116	1.112	1.108	1.104	1.101	1.097	1.094
25–335	1.095	1.089	1.084	1.079	1.075	1.070	1.066	1.062	1.058	1.055	1.051	1.048	1.045	1.042	1.039	1.037
30–330	1.015	1.010	1.006	1.002	0.999	0.995	0.992	0.989	0.986	0.983	0.980	0.977	0.975	0.973	0.970	0.968
35–325	0.924	0.920	0.917	0.914	0.912	0.909	0.907	0.904	0.902	0.900	0.898	0.896	0.895	0.893	0.891	0.890
40–320	0.823	0.821	0.819	0.817	0.815	0.814	0.813	0.811	0.810	0.809	0.808	0.807	0.806	0.805	0.804	0.803
45–315	0.713	0.713	0.712	0.712	0.712	0.711	0.711	0.711	0.710	0.710	0.710	0.710	0.710	0.709	0.709	0.709
50–310	0.598	0.599	0.600	0.601	0.602	0.603	0.603	0.604	0.605	0.606	0.606	0.607	0.608	0.608	0.609	0.610
55–305	0.479	0.481	0.484	0.486	0.488	0.490	0.492	0.494	0.496	0.497	0.499	0.500	0.502	0.503	0.505	0.506
60–300	0.357	0.361	0.365	0.369	0.372	0.375	0.378	0.381	0.384	0.386	0.389	0.391	0.394	0.396	0.398	0.400
65–295	0.236	0.242	0.247	0.251	0.256	0.260	0.264	0.268	0.272	0.275	0.279	0.282	0.285	0.288	0.291	0.293
70–290	0.118	0.124	0.130	0.136	0.142	0.147	0.152	0.156	0.161	0.165	0.169	0.173	0.177	0.180	0.184	0.187
75–285	0.003	0.011	0.018	0.024	0.031	0.037	0.043	0.048	0.053	0.058	0.063	0.067	0.071	0.075	0.079	0.083
80–280	-0.105	-0.097	-0.089	-0.082	-0.075	-0.068	-0.062	-0.056	-0.051	-0.045	-0.040	-0.035	-0.030	-0.026	-0.022	-0.018
85–275	-0.206	-0.197	-0.189	-0.181	-0.174	-0.167	-0.160	-0.154	-0.148	-0.143	-0.137	-0.132	-0.127	-0.123	-0.118	-0.174
90–270	-0.298	-0.289	-0.281	-0.273	-0.265	-0.258	-0.252	-0.245	-0.239	-0.233	-0.228	-0.223	-0.218	-0.213	-0.209	-0.204
95–265	-0.381	-0.372	-0.363	-0.356	-0.348	-0.341	-0.335	-0.328	-0.323	-0.317	-0.312	-0.306	-0.301	-0.297	-0.293	-0.288
100–260	-0.453	-0.444	-0.437	-0.429	-0.422	-0.416	-0.409	-0.403	-0.398	-0.392	-0.387	-0.382	-0.378	-0.373	-0.369	-0.365
105–255	-0.515	-0.507	-0.500	-0.493	-0.487	-0.481	-0.475	-0.470	-0.465	-0.460	-0.455	-0.451	-0.446	-0.442	-0.439	-0.435
110–250	-0.567	-0.560	-0.554	-0.548	-0.543	-0.537	-0.532	-0.528	-0.523	-0.519	-0.515	-0.511	-0.507	-0.504	-0.500	-0.497
115–245	-0.609	-0.604	-0.599	-0.594	-0.589	-0.585	-0.581	-0.577	-0.574	-0.570	-0.567	-0.564	-0.561	-0.558	-0.555	-0.552
120–240	-0.643	-0.639	-0.635	-0.631	-0.628	-0.625	-0.622	-0.619	-0.616	-0.614	-0.611	-0.609	-0.606	-0.604	-0.602	-0.600
125–235	-0.669	-0.666	-0.664	-0.661	-0.659	-0.657	-0.655	-0.653	-0.652	-0.650	-0.648	-0.647	-0.645	-0.644	-0.642	-0.641
130–230	-0.688	-0.687	-0.686	-0.685	-0.684	-0.683	-0.682	-0.681	-0.681	-0.680	-0.679	-0.679	-0.678	-0.677	-0.677	-0.676
135–225	-0.701	-0.701	-0.702	-0.702	-0.703	-0.703	-0.703	-0.704	-0.704	-0.704	-0.704	-0.705	-0.705	-0.705	-0.705	-0.705
140–220	-0.710	-0.712	-0.713	-0.715	-0.717	-0.718	-0.720	-0.721	-0.722	-0.723	-0.724	-0.725	-0.726	-0.727	-0.728	-0.729
145–215	-0.715	-0.718	-0.721	-0.724	-0.727	-0.729	-0.732	-0.734	-0.736	-0.738	-0.740	-0.742	-0.744	-0.745	-0.747	-0.749
150–210	-0.717	-0.722	-0.726	-0.730	-0.734	-0.737	-0.740	-0.744	-0.747	-0.749	-0.752	-0.755	-0.757	-0.760	-0.762	-0.764
155–205	-0.718	-0.723	-0.729	-0.733	-0.738	-0.742	-0.747	-0.751	-0.754	-0.758	-0.761	-0.765	-0.768	-0.771	-0.773	-0.776
160–200	-0.717	-0.724	-0.730	-0.735	-0.741	-0.746	-0.751	-0.755	-0.760	-0.764	-0.768	-0.772	-0.775	-0.779	-0.782	-0.785
165–195	-0.716	-0.723	-0.730	-0.736	-0.742	-0.748	-0.753	-0.759	-0.763	-0.768	-0.772	-0.777	-0.781	-0.785	-0.788	-0.792
170–190	-0.715	-0.723	-0.730	-0.737	-0.743	-0.749	-0.755	-0.761	-0.766	-0.771	-0.776	-0.780	-0.785	-0.789	-0.793	-0.797
175–185	-0.715	-0.722	-0.730	-0.737	-0.744	-0.750	-0.756	-0.762	-0.767	-0.772	-0.777	-0.782	-0.787	-0.791	-0.795	-0.799
180–180	-0.714	-0.722	-0.730	-0.737	-0.744	-0.750	-0.756	-0.762	-0.767	-0.773	-0.778	-0.783	-0.787	-0.792	-0.796	-0.800

Slider-Crank Mechanism

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IN determining the torque caused by the inertia force, the question of a proper division of the weight of the connecting rod between rotating and reciprocating weight arises. For this purpose it is generally assumed that the weight of the rod is concentrated at the piston pin and the crankpin. Letting l_1 be the distance from the center of the crankpin to the center of gravity of the rod, l_2 the distance from the center of gravity to the center of the piston pin, and W_r the weight of the complete rod, the concentrated rotating weight will be $W_r l_2/l$ and the reciprocating, $W_r l_1/l$. Letting W_p be the weight of the complete piston and pin and g the acceleration of gravity

$$\frac{W_2}{g} = \frac{W_f + W_r\, l_1/l}{g}$$

This will be the mass the acceleration of which in the direction of stroke causes the inertia torque. By the same reasoning that was used previously this ineritia torque is

$$M_I = \frac{-W_2\, arF_2}{g}$$

Substituting for a its value is

$$a = r\omega^2 F_3$$

$$M_I = -\frac{W_2\, r^2\, W^2\, F_4}{g}$$

where $F_4 = F_2\, F_3$

Substituting $w = \frac{2\pi n}{60}$

$$T_I = -F_4 \frac{W}{g}\, r^2 \left(\frac{2\pi n}{60}\right)^2$$

wherein

W=weight of reciprocating parts in lb.
r=crank radius
T_1=inertia torque
F_4=factor from table below

From 0 to 180 deg., the signs are as shown in the table, from 180 to 360 deg. the signs are opposite.

With r in in. and g=386 in. per. sec. per sec., T_I will be in lb.-in.

With r in ft. and g = 32.2 ft. per sec. per sec., T_I will be in lb.-ft.

(See article, p. 341-344 for further details and correction factors to be applied.)

Table IV—Factors F_4 For Determining Inertia Torque

Crank Angle Degrees	RATIO OF LENGTH OF ROD TO LENGTH OF CRANK															
	3.5	3.6	3.7	3.8	3.9	4.0	4.1	4.2	4.3	4.4	4.5	4.6	4.7	4.8	4.9	5.0
0–360	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
5–355	0.143	0.141	0.140	0.138	0.137	0.135	0.134	0.133	0.132	0.130	0.129	0.128	0.127	0.126	0.126	0.125
10–350	0.279	0.276	0.273	0.270	0.267	0.264	0.262	0.259	0.257	0.255	0.253	0.251	0.249	0.247	0.246	0.244
15–345	0.402	0.397	0.393	0.388	0.384	0.381	0.377	0.374	0.371	0.367	0.365	0.362	0.359	0.357	0.354	0.352
20–340	0.505	0.499	0.494	0.488	0.483	0.479	0.475	0.471	0.467	0.463	0.460	0.457	0.453	0.450	0.448	0.445
25–335	0.583	0.577	0.571	0.566	0.560	0.555	0.551	0.546	0.542	0.538	0.534	0.531	0.527	0.524	0.521	0.518
30–330	0.634	0.628	0.622	0.616	0.611	0.606	0.601	0.597	0.593	0.589	0.585	0.581	0.578	0.575	0.571	0.568
35–325	0.656	0.650	0.644	0.639	0.634	0.629	0.625	0.621	0.617	0.613	0.610	0.606	0.603	0.600	0.597	0.595
40–320	0.646	0.642	0.637	0.633	0.629	0.625	0.621	0.618	0.615	0.611	0.609	0.606	0.603	0.601	0.598	0.596
45–315	0.608	0.605	0.602	0.599	0.596	0.593	0.591	0.588	0.586	0.584	0.582	0.580	0.578	0.576	0.575	0.573
50–310	0.544	0.543	0.541	0.540	0.538	0.537	0.536	0.535	0.534	0.533	0.532	0.531	0.530	0.529	0.529	0.528
55–305	0.458	0.459	0.459	0.459	0.460	0.460	0.460	0.461	0.461	0.461	0.462	0.462	0.462	0.462	0.463	0.463
60–300	0.355	0.358	0.360	0.362	0.364	0.366	0.368	0.370	0.372	0.373	0.375	0.377	0.378	0.379	0.380	0.382
65–295	0.241	0.246	0.250	0.254	0.258	0.261	0.265	0.268	0.271	0.274	0.277	0.379	0.282	0.284	0.286	0.289
70–290	0.122	0.128	0.134	0.140	0.145	0.150	0.155	0.159	0.163	0.168	0.171	0.175	0.179	0.182	0.185	0.188
75–285	0.003	0.011	0.018	0.025	0.032	0.038	0.044	0.049	0.054	0.059	0.064	0.068	0.073	0.077	0.081	0.084
80–280	-0.109	-0.100	-0.092	-0.084	-0.077	-0.070	-0.064	-0.058	-0.052	-0.046	-0.041	-0.036	-0.031	-0.027	-0.022	-0.018
85–275	-0.211	-0.202	-0.193	-0.185	-0.177	-0.170	-0.163	-0.157	-0.151	-0.145	-0.139	-0.134	-0.129	-0.124	-0.120	-0.115
90–270	-0.298	-0.289	-0.281	-0.273	-0.265	-0.258	-0.252	-0.245	-0.239	-0.233	-0.228	-0.223	-0.218	-0.213	-0.209	-0.204
95–265	-0.369	-0.361	-0.353	-0.346	-0.339	-0.332	-0.326	-0.320	-0.315	-0.309	-0.304	-0.299	-0.295	-0.290	-0.286	-0.282
100–260	-0.423	-0.416	-0.409	-0.403	-0.397	-0.391	-0.386	-0.380	-0.376	-0.371	-0.366	-0.362	-0.358	-0.354	-0.350	-0.347
105–255	-0.459	-0.453	-0.448	-0.443	-0.438	-0.434	-0.429	-0.425	-0.421	-0.417	-0.414	-0.410	-0.407	-0.404	-0.401	-0.398
110–250	-0.478	-0.474	-0.471	-0.467	-0.464	-0.461	-0.457	-0.454	-0.452	-0.449	-0.446	-0.444	-0.441	-0.439	-0.437	-0.435
115–245	-0.483	-0.481	-0.479	-0.477	-0.475	-0.473	-0.471	-0.469	-0.468	-0.466	-0.464	-0.463	-0.462	-0.460	-0.459	-0.457
120–240	-0.475	-0.474	-0.474	-0.473	-0.473	-0.472	-0.471	-0.471	-0.470	-0.470	-0.469	-0.469	-0.468	-0.468	-0.467	-0.467
125–235	-0.455	-0.456	-0.457	-0.458	-0.459	-0.460	-0.460	-0.461	-0.461	-0.462	-0.462	-0.463	-0.463	-0.463	-0.464	-0.464
130–230	-0.428	-0.430	-0.432	-0.434	-0.436	-0.438	-0.439	-0.441	-0.442	-0.444	-0.445	-0.446	-0.447	-0.448	-0.449	-0.450
135–225	-0.393	-0.397	-0.400	-0.403	-0.405	-0.408	-0.410	-0.413	-0.415	-0.417	-0.419	-0.421	-0.422	-0.424	-0.426	-0.427
140–220	-0.355	-0.359	-0.362	-0.366	-0.369	-0.372	-0.375	-0.378	-0.381	-0.383	-0.386	-0.388	-0.390	-0.392	-0.394	-0.396
145–215	-0.313	-0.317	-0.321	-0.325	-0.328	-0.332	-0.335	-0.338	-0.341	-0.344	-0.347	-0.349	-0.352	-0.354	-0.356	-0.359
150–210	-0.269	-0.273	-0.277	-0.281	-0.285	-0.288	-0.291	-0.295	-0.298	-0.300	-0.303	-0.306	-0.309	-0.311	-0.313	-0.316
155–205	-0.224	-0.228	-0.232	-0.236	-0.239	-0.242	-0.245	-0.248	-0.251	-0.254	-0.257	-0.259	-0.262	-0.264	-0.266	-0.268
160–200	-0.179	-0.183	-0.186	-0.189	-0.192	-0.195	-0.198	-0.200	-0.203	-0.205	-0.208	-0.210	-0.212	-0.214	-0.216	-0.218
165–195	-0.134	-0.137	-0.140	-0.142	-0.145	-0.147	-0.149	-0.151	-0.153	-0.155	-0.157	-0.159	-0.161	-0.162	-0.164	-0.165
170–190	-0.089	-0.091	-0.093	-0.095	-0.097	-0.098	-0.100	-0.101	-0.103	-0.104	-0.105	-0.106	-0.108	-0.109	-0.110	-0.111
175–185	-0.045	-0.046	-0.047	-0.047	-0.048	-0.049	-0.050	-0.051	-0.051	-0.052	-0.053	-0.053	-0.054	-0.055	-0.055	-0.056
180–180	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000